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ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

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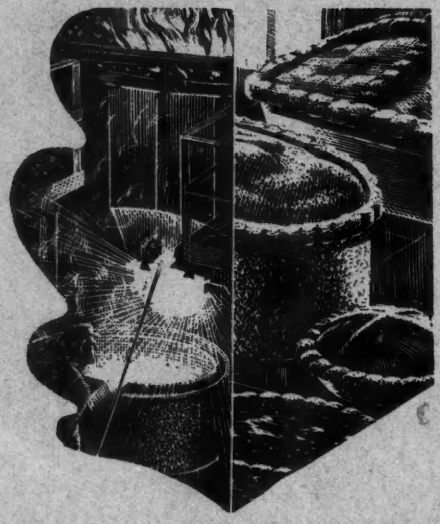

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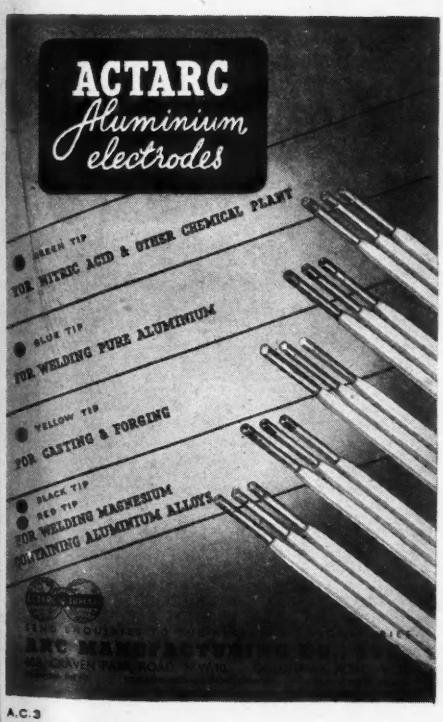

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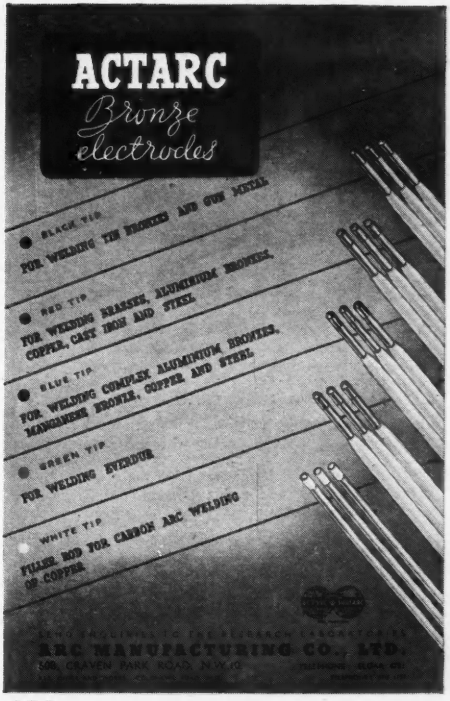
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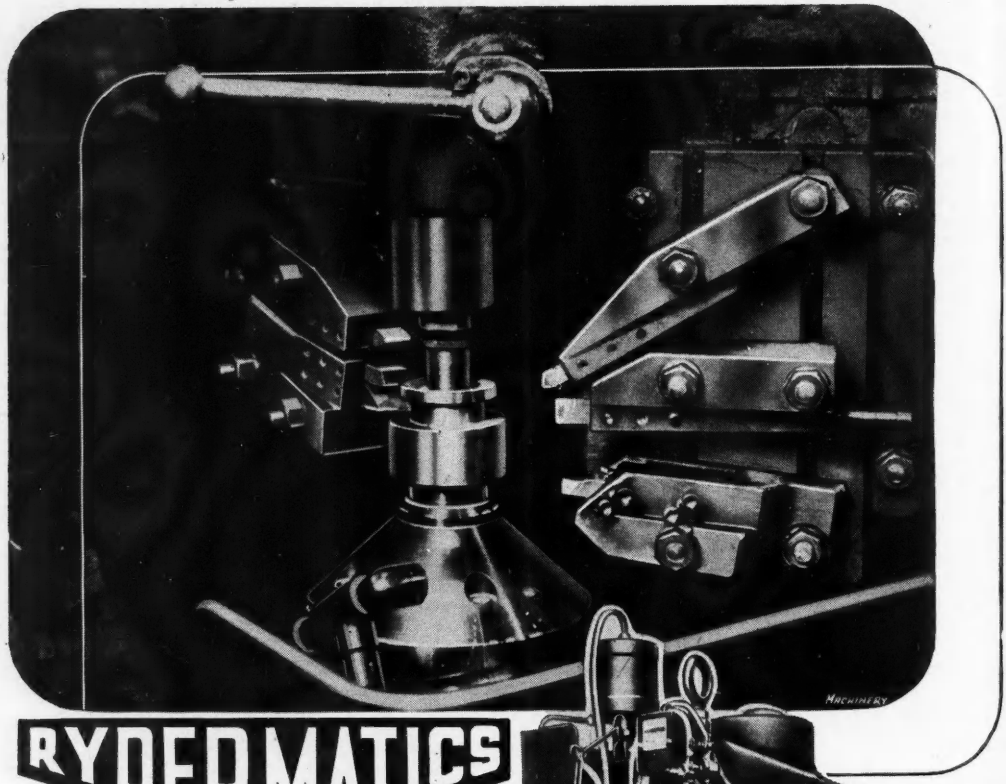

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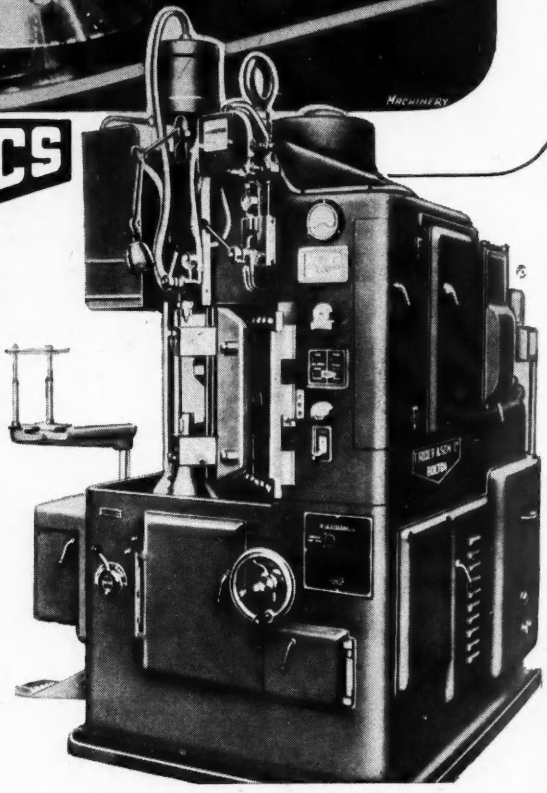

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The Editor and Publishers of "The Engineers' Digest" wish their Subscribers, Advertisers and Collaborators, the Compliments of the Season—a Prosperous and Successful New Year.

GERMAN DIE CASTING PLANTS

From United States Office of War Information Service.

THE findings of the investigators into German die casting plants have proved that the German industry is chiefly confined to two areas, namely, Berlin and Stuttgart-Munich; as it was not possible to inspect the Berlin district, all subsequent general remarks should be considered as referring to the latter area.

The plant is mainly of the Polak type, and with one exception is of an age which gives uneconomical production. An outstanding feature was the absence of heat temperature control in most plants in the holding pots.

While the workmanship was good, it was uninspired and not up to U.S.A. or British standards. Due to shortage of alloying elements, inferior steels were used. Core pulling was conventional—angle pins and hydraulic.

Finishing equipment was very ordinary and far below U.S.A. and British standards. X-ray apparatus for inspection was struck on only two occasions and then it was doubtful whether the apparatus would be suitable for all castings and materials. The only inspection appeared to be of visual nature, even weight testing was not employed.

Production rates were low in comparison with U.S.A. and Great Britain. The output per machine also seemed low.

Die cast materials were confined to three: namely, Magnesium, Aluminium and Zinc. The results on magnesium were good, but no better than in U.S.A. or Great Britain, and the reasons for Germany's fairly high production in this material was probably controlled by supply.

Nothing in the way of new alloys, die steels or hydraulic mediums was seen. Hydraulic systems were usually water and soluble oil.

German die casters and the American investigators had one thing in common. That was a desire to see the plants operating again. The Germans wanted to let by-gones be by-gones and start a chummy little German-American Die Casting Bund. One of the first aims of such a club would be the importation of some up-to-date die casting machines from America.

It was difficult to make a really good appraisal of a plant that is not in operation, but the investigators were less handicapped than would be expected because the die casting machines were nearly all of the old type rocking air gooseneck or Polak, Nos. 408, 600, 2255, or copies of them. One outstanding exception was Mahle vertical air gooseneck type. This machine looked rather business like, but with most glaring inconsistencies. The sample castings from this machine were large and complicated and of excellent quality. As a matter of fact, the final casting results obtained by the German die casters seemed in general to be superior to what one might expect from inspecting the die casting machines. It was the familiar story of cheap labour and slave labour and good organization at the top.

As far as the large Mahle verticle machine was concerned, the labour not only had to be cheap, but expendable. The valving of the air gooseneck was submerged in the metal with no provision against chips or foreign matter lodging on the valve seat. An open valve at the time of the shot meant a blow-out of all the metal in the metal pot. The chief engineer of the main Mahle plant admitted that the machine was most dangerous to personnel and said they had had many accidents and some of them fatal. He had eventually designed and put in operation a series of electrical switches on the machine and on the door of the anti-room so that the personnel had to "run quick in the office" before the machine was shot. Such a poorly designed machine was not up to the touted German efficiency.

Mahle executives claimed to do 65 per cent of all the magnesium die casting in Germany and 35 per cent of the aluminium die casting and a substantial tonnage in Zinc alloy. Their Magnesium casting was also done in the large verticle air gooseneck machines of the Mahle displacement plunger type.

The tiebars on the large Mahle Verticle were only approximately 3½ inches in diameter, which seemed altogether too small for the sizes of castings which had been made on it. Much of the pressure load, however, was taken by auxiliary hydraulic wedge locks on the dies. These locks were placed on opposite sides and extended over about one-half the width or length of the die. Provision could be made to place the die locks on all four sides of the die if necessary. The central hydraulic system seemed adequate to handle numerous auxiliary die locks and core pulling devices. The core pulling hydraulic cylinders were not made a part of the dies, but were standard extras to be applied as needed and in any sizes needed. The Mahle store room was very well supplied with these extra cylinder assemblies.

Water and soluble oil seemed to be the standard hydraulic fluid. Polak or Mullor systems were popular.

The melt-down rooms were interesting from several standpoints. First of all, the larger plants were always equipped with duplicate melt-down capacity—one set for electric firing and the other for gas. If the "Gas House" got blown away and the electric plant didn't, a switch-over could be accomplished without much time loss. If both got knocked out temporarily, the plants improvised as best they could with coke firing in the gas furnaces until the "juice" came back. Quick repairs to damaged plants seemed to be a very strong point in the German planning. The race between the repair gang in Germany and our bombers was, of course, won by the bombers, but at times, or for a time at least, the race was a close one. The ventilating system for Magnesium castings, if any, was inadequate, and the plant personnel was said to suffer a great deal from fluxing fumes and SO_2 gas. Furnace equipment seemed to be excellent, especially the new reverberatory electric furnaces.

Generally, the Germans "had nothing on us," and, therefore, whatever comfort such a negative report may give, may be enjoyed to the fullest.

EXCEPTIONALLY WIDE BAND-PASS WAVE FILTERS THROUGH ARTIFICIALLY BRED CRYSTALS.

By W. BANTLE, B. MATTHIAS and P. SCHERRER. (From *Schweizer Archiv*, Vol. 11, No. 6, June, 1945, pp. 161-164).

THE use of piezoelectric crystals in electric wave filters proved to be one of the most successful of their numerous technical applications. If an alternating electric field is applied to a conveniently cut slab of a piezoelectric crystal, the latter is excited to mechanical oscillations. If the frequency of the field is equal to the mechanical resonant frequency of the slab, the amplitude of the mechanical oscillations becomes a maximum. A condenser in which the crystal slab forms the dielectric shows, therefore, the phenomenon of an electric series and parallel resonance. The equivalent electric circuit of a crystal used as a dielectric in a condenser is shown in Fig. 1. It is simply a series resonant circuit containing

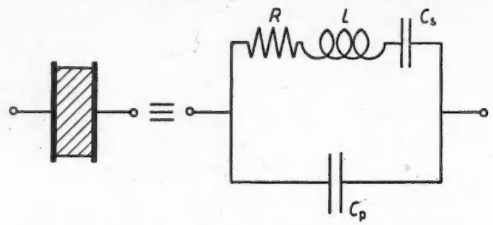


Fig. 1. **Equivalent electric circuit of an oscillator crystal.**

resistance, capacitance, and inductance paralleled by the capacitance of the holder. The impedance of this circuit (disregarding the ohmic resistance) at various frequencies is shown in Fig. 2. At low frequencies the crystal behaves like a capacitance; at the resonance frequency ω_R, the reactance is zero; thereafter the reactance assumes inductive values and increases with the frequency until it reaches infinity at a certain frequency ω_A, called the antiresonance frequency. At frequencies above ω_A the reactance becomes capacitive again. The attenuation of a mechanical oscillating system and in particular of an oscillating crystal is far below the attenuation obtainable with electric oscillating circuits consisting of coils and condensers. A measure of the attenuation of a crystal resonance circuit is the so-called " Gütefaktor "

$$Q = \frac{1}{R}\sqrt{\frac{L}{C_s}} \quad .. \quad .. \quad .. \quad (1)$$

The values of the elements of the equivalent electric circuit are determined by the size of the slab, by its physical properties (density, Young's modulus, piezoelectric constant, etc.), and the crystallographic cut by which it was produced. If a definite resonance frequency is required and a certain type of crystal is to be used, the individual characteristic data of the equivalent circuit can vary only within very narrow limits. For instance, the *maximum* value $\frac{C_s}{C_p}$, which determines the distance between resonance and anti-resonance frequency is solely dependent on the physical properties of the crystal and not on its dimensions. For a transversely oscillating crystal this value is :

$$\frac{C_s}{C_p} = \frac{16}{\pi} \cdot \frac{d^2E}{\epsilon} \quad .. \quad .. \quad .. \quad (2)$$

where d = piezoelectric constant, E = Young's modulus, and ϵ = dielectric constant. This formula provides also a measure of the frequency range in which the crystal behaves like an inductance, since the following equation is also valid :

$$\frac{C_s}{C_p} \cong 2 \cdot \frac{\omega_A - \omega_R}{\omega_R} \quad .. \quad .. \quad (3)$$

For a given ω_R, there is also a lower limit for L which, if a transversely oscillating crystal is used, is determined by the smallest thickness of the slab which can be obtained in practice.

This shows that the values of the elements of the equivalent circuit are strictly limited, a fact which considerably restricts the use of piezoelectric crystals in wave filters.

Up to the present, the crystals used for wave filters have been almost exclusively of quartz. For this material the upper limit of $\frac{C_s}{C_p}$ is 8 per mil., while the lowest obtainable values for L are between 20 and 30 Henry. This means that the band width of quartz filters cannot be made wider than 13 per cent of the mean transmitted frequency; but even for this value the characteristic impedance of the circuit reaches values which render it useless for all practical purposes. This fact, however, is not obvious and will be explained in the following.

The most generally used crystal filter is of the Cauer's bridge type, in which two pairs of equal impedances are used. It is, therefore, a symmetrical four-terminal network with reversible input and output

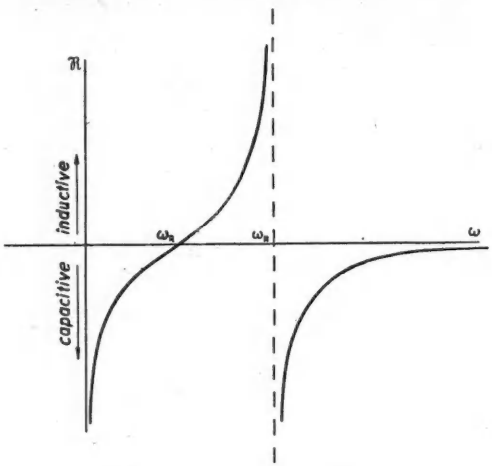


Fig. 2. **Reactance of a crystal.**

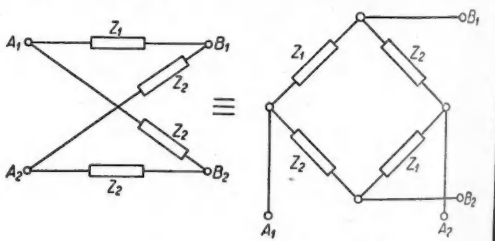


Fig. 3. **Circuit diagram of a filter.**

terminals. At frequency ranges for which the impedances Z_1 and Z_2 are nearly equal, the bridge is largely in equilibrium and the transmission will be blocked. If, at certain frequencies, Z_1 and Z_2 are exactly equal, the attenuation will be infinite (attentuation peaks). Now it is possible to arrange that, for a definite frequency range, the impedances Z_1 and Z_2 differ greatly from one another, for instance, by giving Z_1 inductive and at the same time Z_2 capacitive properties. Under such conditions the filter will not be in equilibrium and will transmit the frequency range in question.

It is therefore obvious that the band width obtainable with a filter of this type is strictly limited, if the impedances are formed by crystals with a characteristic as shown in Fig. 2, which have a capacitive behaviour over the whole frequency range except for the short range between resonance and antiresonance. If, however, two pairs of crystals are used, where each pair consists of two similar crystals, and the resonance frequency of the one pair coincides with the antiresonance frequency of the other, the width of the transmission band of the filter will be twice the distance between resonance and antiresonance of one individual crystal. This means that, according to equation (3) the maximum relative band width obtainable with a filter consisting entirely of quartz crystals is 8 per mil. Such a band width is, however, totally inadequate for multiple channel carrier telephony between 30 and 200 kc/s.

The relative band width of such a filter can be increased if induction coils L^* are connected in series with the crystals, which means that the " all-crystal " character of the filter is given up. This has, amongst others, the disadvantage that the characteristics of the equivalent two-terminal networks of the new element have two zero points ω_1 and ω_2 (Fig. 4). The actual

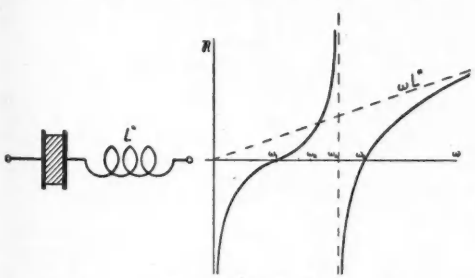


Fig. 4. **Reactance of a crystal with series inductance.**

values for the elements of the two pairs of elements must be chosen so that within the transmission band of the filter the inductive range of the one pair corresponds to the capacitive range of the other and vice versa. In the elimination range both pairs must be either inductive or capacitive (Fig. 5). The resultant width of the transmission band is from ω_1 to ω_4. The attenuation peaks are at $\omega_{\infty 1}$ and $\omega_{\infty 2}$.

But also by using additional inductances, the practical band width of a quartz crystal filter cannot be increased indefinitely. Quartz filters, usable in practice, can be made only up to a band width of 6–7 per cent as at higher values the adjustability rapidly decreases. This is due to the characteristic impedance of a crystal filter with induction coils. This characteristic impedance $Z = \sqrt{Z_1 Z_2}$ changes considerably with the frequency; it is zero at the fronts of the transmission band (neglecting the ohmic resistance), as at each front of the band one impedance is zero while the other is not infinite. It increases towards the centre of the transmission band and reaches a maximum value of $Z_{max} = (\omega_4 - \omega_1) L^*$. If each filter could be connected to a load whose characteristic impedance would correspond to that of the filter itself at each individual frequency, the transmission diagram of the filter would have a practically rectangular shape due to the very high value of Q (see equation 1). The unfavourable effect of the additional inductances can be avoided, if all four have the same value and if they are connected outside the bridge according to Bartlett's rule of equivalent four-terminal networks; by this measure they produce only a negligible frequency-independent additional attenuation. In practice, however, the load connected to the filter does not change with the frequency and a mean characteristic impedance must be assumed; consequently, if the width of the band is increased, the absolute value of the characteristic impedance increases according to the above equation. The practical adaptability of quartz filters is, therefore, possible only up to a relative band width of 6 to 7 per cent. Even an increase to 9 to 10 per cent, which would be desirable for high quality sound transmission at 100 kc/s., encounters extraordinary difficulties. In principle (disregarding the difficulties of adjustment), however, quartz filters with additional inductances can be built up to a relative band width of 13 per cent. Beyond this value the elimination attenuation cannot be effectively controlled, which renders the filter impracticable for technical purposes.

In a search for filters allowing a band width beyond the limits analysed in the above, the suitability for use in filters, of artificially bred crystals has been investigated. First to be considered are all phosphates of potassium and of ammonium (KH_2PO_4 and $NH_4H_2PO_4$) which show much more favourable values for $\frac{C_s}{C_p}$ and L than quartz crystals, while the " Gütefaktor " of their equivalent circuit is of the same order as that for quartz.

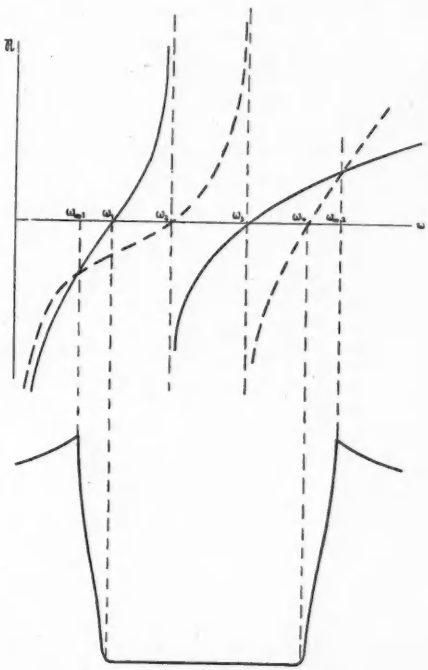


Fig. 5. **Reactance and attenuation.**

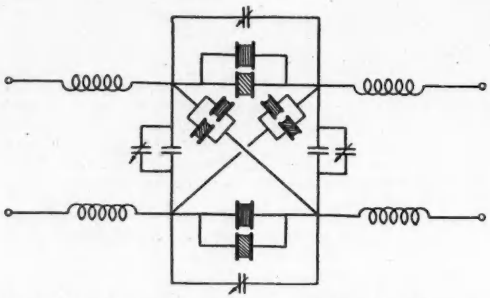


Fig. 6. **Band-pass filter for very wide bands with two different types of crystals.**

The two phosphates mentioned make it possible for the first time to build band filters also for relative band widths far exceeding those obtainable with quartz crystals. For instance, the maximum band width obtainable with a filter of the type described above is 33 per cent against 13 per cent if quartz crystals are used. An additional advantage is that, owing to the more favourable values for L in the equivalent circuit, the values of the additional inductances can be kept so low that no appreciable adjustment difficulties occur.

The question arises whether it is possible to obtain, for a frequency range between 30 and 100 kc/s., filters for a band width of 9 to 10 kc/s. whose characteristic impedance would not exceed the limits set for practical application, i.e., 600 ohms. This condition can actually be fulfilled by using crystals of potassium and ammonium phosphate simultaneously.

For the first time filters of the Cauer's bridge type were calculated and produced which had two crystals of different material connected in parallel in each branch (Fig. 6). This arrangement provides the possibility of varying the values of the elements of the equivalent electric circuit of this four-terminal network within much wider limits than has hitherto been possible, and values can be obtained which ensure the required band width of 9–10 kc/s. while at the same time the necessary adaptability is maintained and the extraordinary slope of the band fronts of 5 Np/100 kc/s. is reached. To ensure an adequate elimination attenuation, two filters were connected in series; for this arrangement the attenuation characteristic shown in Fig. 7 was obtained.

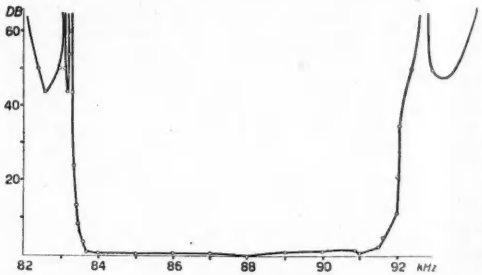


Fig. 7. **Attenuation of two band-pass filters for very wide bands connected in series.**

CASCADE ASYNCHRONOUS GENERATOR.

By P. A. SVIRIDENKO. (From *Izvestia Akad. Nauk. U.S.S.R.*, Otdel. Tech. Nauk, 1945, Nos. 1 to 2, pp. 94-107).

THE electric circuit shown in Fig. 1 outlines a system employed for the recuperation of electric energy in the testing of aero engines. In this diagram the aero engine AM is coupled with two identical induction machines ZM–1 and ZM–2, each rated at 390 kW. at 1500 r.p.m. and laid out for a stator voltage of 6,000 volts. Starting of the set is either effected by the aero engine or by running the set as motor.

When testing the aero engines with gas as fuel at low engine speeds ranging from 750 to 1,400 r.p.m., the induction machines are operated in cascade, the switches MB–1, MB–3 and KT–1 being closed and the switches MB–2, KT–2, and KT–3 being open. The negative slip can be controlled within wide limits by (a) variation of the resistor R–3 arranged in parallel with the stator coil of the second machine and connected to the secondary winding of the step down transformer, and (b) by controlling the fuel supply to the engine. After the engine speed has reached 1400 r.p.m., the switches MB, MB–2, KT–2, and KT–3 are closed and the switches MB–3 and KT–1 are opened, switch MB–1 remaining closed. The engine speed is then raised to 1800 r.p.m. The resistors R–1 and R–2 are remotely controlled from the operating panel. Idling of the aero engines may be realised by operating the induction machines as cascade converters over the speed range of 120–750 r.p.m.

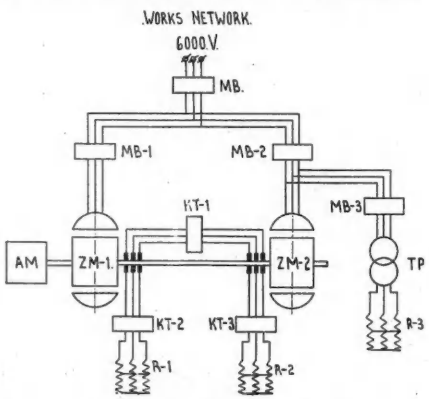


Fig. 1. **Circuit of cascade-generator set for testing aero engines.**

A M = aero engine Z M—1, Z M—2, = induction machines of 390 kW. capacity each; R = resistor; TP = transformer,

The losses in the rotor and in the resistor are equal to the power transmitted across the air gap multiplied by the slip, that is:

$$\Delta P_{rot} = P_{air\ gap} \cdot S = (P_2 + \Delta P_{st})\, S \quad .. \quad (1)$$

where P_2 is the power returned into the network. The total losses are

$$\Delta P = P_{rot} + \Delta P_{st} + \Delta P_{mech} + \Delta P_{add} \quad .. \quad (2)$$

where ΔP_{st} are the core and copper losses of the stator. These are given by

$$\Delta P_{st} = \Delta P_{iron.\ st} + P_{copper\ st.} \left[\frac{I_1}{I_{1n}}\right]^2 \quad .. \quad (3)$$

where (I_{1n}) denotes the nominal current. The windage losses increase with the third power of the speed and the friction losses with the 1·5th power, so that

$$\Delta P_{mech} = \Delta P_{windage} \left[\frac{n}{n_s}\right]^3 + \Delta P_{friction} \left[\frac{n}{n_s}\right]^{1.5} \quad (4)$$

The additional losses are proportional to the square of the speed and to the square of the stator current, so that

$$\Delta P_{add} = \Delta P_{add} \left(\frac{n}{n_s}\right)^2 \left(\frac{I_1}{I_{1n}}\right)^2 \quad .. \quad (5)$$

Formula (2) can therefore be written

$$P_2 = P_1 \frac{n_s}{n} - \Delta P_{windage} \left(\frac{n}{n_s}\right)^2 - \Delta P_{friction} \left(\frac{n}{n_s}\right)^{0.5} - \Delta P_{add} \left(\frac{n}{n_s}\right) K_y^2 - \Delta P_{copper\ st}\ K_y^2 - \Delta P_{iron\ st} \text{ where } K_y = \frac{I_1}{I_{1n}} \quad .. \quad .. \quad (6)$$

This equation shows that, when operating as generator, the induction machine has a somewhat lower efficiency than when operating as motor, assuming identical absolute slips. Experimental data confirm that the output delivered into the network at different speeds and shaft powers may be obtained from equation (6) with sufficient accuracy.

In Fig. 2 are shown the characteristics obtained with an actual installation. In the testing of aero engines it is required that the load imposed upon the engine should have the same characteristics as those produced by the airscrew. The larger part, P_2, of this load is returned into the network. Return of energy into the network takes place at engine speeds ranging from 1130–2,700 r.p.m. At speeds between 1,130–2,120 r.p.m. the machines operate as cascade generators, while they operate in parallel over the speed range of 2,250–2,700 r.p.m.

For a test run of 2 h. 15 min. the electrical energy returned into the network amounts to 71 per cent of the energy transmitted to the shaft, that is, to some 700 kWh. Tests have shown that the load characteristic imposed upon the aero engine can be changed at will. The range of possible shaft loads is given in Fig. 2 by the limiting curve P_1'.

Referring to the equivalent circuit given in Fig. 3 it is seen that the cascade rotor current can be expressed by

$$I_2 = \frac{U}{\left(\frac{R_2}{S_1} + R_1\right) + \frac{\frac{R_3}{S_2} X_{m_2}^2}{\left(\frac{R_3}{S_2}\right)^2 + X_{III}^2} + j\left[X_{II} - \frac{X_{III} X_{m_2}^2}{\left(\frac{R_3}{S_2}\right)^2 + X_{III}^2}\right]} \quad .. \quad (7)$$

and the current in the stator of the second machine by

$$I_3 = \frac{U}{-\left[\left(\frac{R_2}{S_1} + R_1\right)\frac{X_{III}}{X_{m_2}} + \frac{R_3}{S_2}\frac{X_{II}}{X_{m_2}}\right] + j\left[\frac{R_3}{S_2}\left(\frac{R_2}{S_1} + R_1\right)\frac{1}{X_{m_2}} - \frac{X_{II} X_{III}}{X_{m_2}} + X_{m_2}\right]} \quad (8)$$

where S_1 is the slip of the first machine which has the synchronous speed n_o. It is

$$S_1 = \frac{n_o - n}{n_o} \quad .. \quad .. \quad .. \quad (9)$$

The slip of the second machine, which has the synchronous speed $n_{ok} = \frac{n_o}{2}$ is defined as

$$S_2 = \frac{n_{ok} - n}{n_{ok}} = 1 - \frac{2n}{n_o} \quad .. \quad .. \quad (10)$$

From formulae (9) and (10) we obtain

$$S_2 = 2\,S_1 - 1 \quad .. \quad .. \quad .. \quad (11)$$

If the speed changes from n_{ok} to n_o, the slip S_1 and the frequency at the rotor terminals of the machines will change within the limits 0·5–0, while the slip and the frequency at the terminals of the stator of the second machine will change within the limits 0 — (—1·0). The torque of the first machine is

$$M_1 = \left(\frac{R_2}{S_1} I_2^2 + \frac{R_3}{S_2} I_3^2\right) \frac{mp}{\omega_1} \quad .. \quad (12)$$

and that of the second machine is

$$M_2 = \frac{R_2}{S_2} I_3^2 \frac{mp}{\omega_1} \quad .. \quad .. \quad .. \quad (13)$$

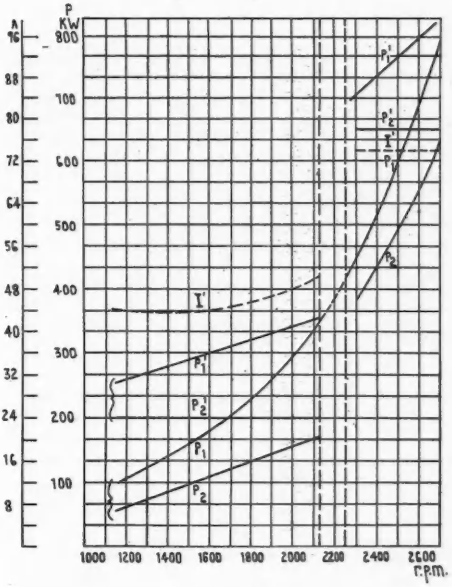


Fig. 2

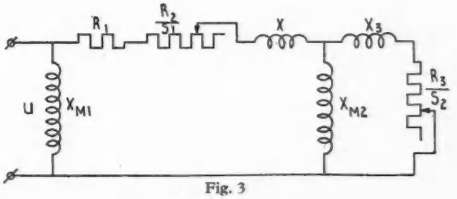


Fig. 3

The resultant torque therefore is

$$M = M_1 + M_2 = \left(\frac{R_2}{S_1} I_2^2 + \frac{2R_3}{S_2} I_3^2 \right) \frac{mp}{\omega_1} \quad (14)$$

In Fig. 4 are shown the torque curves computed from equations (12) and (13) as functions of the speed of the cascade generator for three different resistances in the rotor circuit, namely $R_p = 0$, $R_p = 17\ R_{cm}$, and $R_p = 55\ R_{cm}$. From these data it will be seen that the torque has two maxima, i.e., a generator maximum and a motor maximum. The maximum torque decreases somewhat with increased rotor resistance because of the increase of the equivalent active resistance $\frac{R_2}{S_1}$. (See Fig. 3).

At speeds slightly below synchronous speed, the maximum torque of the generator drops sharply to zero, and stable operation of the cascade generator becomes impossible. From Fig. 4 it is seen that the maximum torque of the cascade generator is two times greater than the maximum turning moment of the engine.

Because of the relatively small reactive resistance of the cascade the current increases when the machine runs idle, while the overload capacity is considerably decreased. The actual overload characteristic could not, however, be ascertained, as the turning moment of the aviation Diesel used in the test did not exceed the torque which the cascade set was able to carry. In Fig. 5 are shown the variations in voltage, current, slip, and electric output of the rotor as functions of the rotational speed for constant output of the cascade generator.

The electric output P_{12} transferred from the rotor of the second machine to the first machine equals the power transmitted across the air gap multiplied by the slip of the first machine, so that

$$P_{12} = P_{\text{air gap}} \cdot S_1 \quad .. \quad .. \quad (15)$$

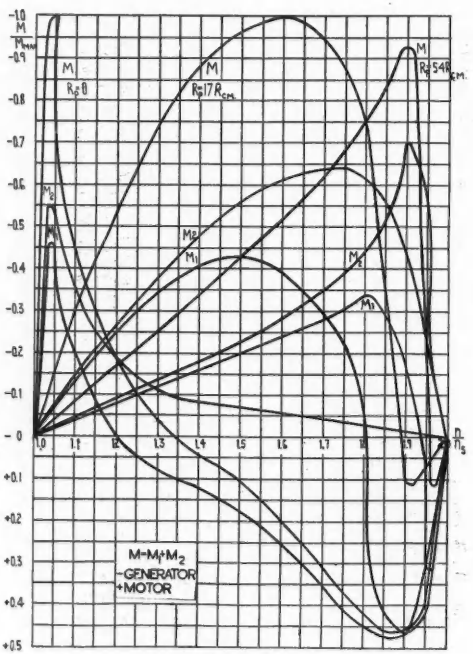


Fig. 4

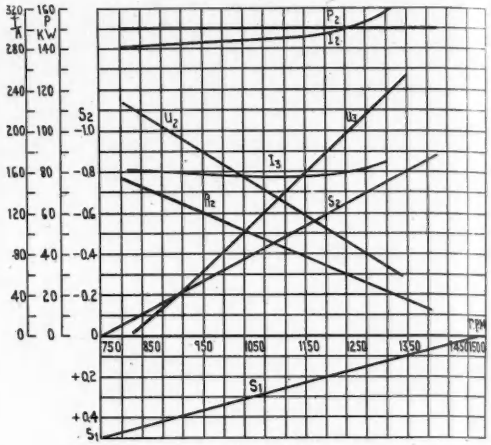


Fig. 5

This capacity is about one-half the output delivered into the network if the speed of the cascade approaches the synchronous speed ($S_1 \approx 0{\cdot}5$). By increasing the speed, the active power in the rotor circuit will decrease according to a straight-line law, and at the limit speed it will be zero. At the same time the losses in the stator and in the resistor of the second machine are

$$P_{st\ 2} = P_{\text{air gap}} \cdot S_2 \quad .. \quad .. \quad (16)$$

In the rotor and stator circuits of the second machine the frequency will change over a wide range, but in the range considered the magnetization of the machine will change but little from the condition prevailing at constant engine-speed. This is explained by the fact that voltage as well as frequency of the cascade circuits vary closely according to the same law, namely,

$$U_2 \approx E_2 \cdot S_1 \qquad U_3 \approx E_3 \cdot S_2$$

The voltage of the stator circuit of the second machine varies from zero to nearly the nominal value. In order to reduce the voltage at the terminals of the resistor, a 6,000/425 volt step-down transformer is interposed. In the primary circuit of this transformer voltage and frequency vary within wide limits. The ratio of primary to secondary current therefore will depend not only upon the parameters of the transformer but also upon the frequency of the primary current.

Considering the external circuit connected to the stator of the second machine we have

$$\frac{I_2}{I_1} = \frac{M}{\sqrt{(M+L)^2 + \frac{R}{\omega_2}}} \quad .. \quad .. \quad (17)$$

where M is the coefficient of the mutual induction of the windings of the transformer, L is the coefficient of self-induction, and R is the resistance of the resistor while ω_2 is the frequency at the primary terminals of the transformer. The variation in the ratio $\frac{I_2}{I_1}$ with the frequency for different resistances and coefficiencts of mutual induction is charted in Fig. 6. Here it will be seen that a decrease in frequency is equivalent to an increase in the ohmic resistance, the influence of the frequency being particularly pronounced in the range of low frequencies. If the slip is small, then the mutual induction of the transformer windings is also very small and hardly any current will flow in the secondary circuit.

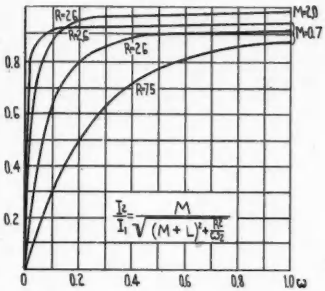


Fig. 6

The frequency limits within which a marked variation in the ratio $\frac{I_2}{I_1}$ obtains, are seen to become greater with increasing ohmic resistance and with decreasing coefficients of mutual induction.

The engine test procedure includes a run during which the engine is turned over cold at varying speeds. In this case power must be taken from the network. According to the test data obtained, the energy consumption will then vary with the speed as follows :

Speed of aero engine (cold), r.p.m.	180	300	500	720	940	1,000	1,100
Power taken from network, kW.	77	80	83	85	84	80	80

FUNDAMENTAL EQUATIONS OF THE CASCADE CIRCUIT.

The theoretical analysis of the various operating conditions of the cascade generator requires the establishment of the fundamental differential equations. In order to be able to establish these equations we replace the three-phase cascade system of the induction machines by a two-phase system. By resorting to this artifice the general equations become free of the trigonometric coefficients so that linear equations with constant coefficients can be established for constant rotor speed. Fig. 7 shows the cascade arrangement of two electrically and mechanically inter-connected induction machines with the stator phases a, b, c, a', b', c', and the rotor phases $1, 2, 3, 1', 2', 3'$. According to Kirchhoff's law the following equations can be established :

For the stator of the first machine

$$\left.\begin{aligned} U_a &= p\Psi_a + R_1 i_a \\ U_b &= p\Psi_b + R_1 i_b \\ U_c &= p\Psi_c + R_1 i_c \end{aligned}\right\} \quad .. \quad .. \quad (18)$$

For the rotor circuits of the two machines

$$\left.\begin{aligned} p(\Psi_1 + \Psi_1') + (r_1 + r_1') i_1 &= 0 \\ p(\Psi_2 + \Psi_2') + (r_1 + r_1') i_2 &= 0 \\ p(\Psi_3 + \Psi_3') + (r_1 + r_1') i_3 &= 0 \end{aligned}\right\} .. \quad (19)$$

For the stator of the second machine

$$\left.\begin{aligned} p\Psi_a' + (R_1' + R_p) i_a' &= 0 \\ p\Psi_b' + (R_1' + R_p) i_b' &= 0 \\ p\Psi_c' + (R_1' + R_p) i_c' &= 0 \end{aligned}\right\} \quad .. \quad (20)$$

where $U_{a,b,c}$ are the voltages of the stator phases of the first machine, $\Psi_{a,b,c}$; $\Psi_{1,2,3}$; $\Psi_{a',b',c'}$; $\Psi_{1',2',3'}$ are the interconnected phases of stator and rotor of the first and second machine ; $i_{a,b,c}$; $i_{1,2,3}$; $i_{a',b',c'}$ are the currents in the stator and rotor phases, R_1, r_1, r_1', R_1' are the ohmic resistances of stator and rotor of the machines and R_p those of the resistor phases, $p = \frac{d}{dt}$ is the differential operator.

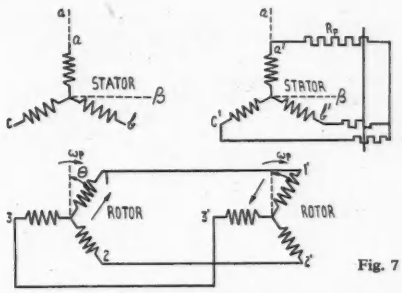


Fig. 7

By transforming these equations into an α, β, 0 system, the following fundamental equations can be obtained

$$\begin{aligned} &(R_1 + L_{01}p) I_{\alpha 1} + 0 + M_{01} pi_\alpha + 0 + 0 + 0 = U_\alpha \\ &0 + (R_1 + L_{01}p) I_{\beta 1} + 0 + M_{01} pi_\beta + 0 + 0 = U_\beta \\ &M_{01}p I_{\alpha 1} + M_{01}[p\Theta] I_{\beta 1} + (r + l_0 p) i_\alpha + l_0 [p\Theta] i_\beta \\ &\quad + M_{02} pI_{\alpha 2} + M_{02}[p\Theta] I_{\beta 2} = 0 \\ &-M_{01}[p\Theta] I_{\alpha 1} + M_{01} pI_{\beta 1} - l_0 [p\Theta] i_\alpha + \quad (21) \\ &\quad + (r + l_0 p) i_\beta - M_{02}[p\Theta] I_{\alpha 2} + M_{02} p I_{\beta 2} = 0 \\ &0 + 0 + M_{02} pi_\alpha + 0 + (R_2 + L_{02} p) I_{\alpha 2} + 0 = 0 \\ &0 + 0 + 0 + M_{02} pi_\beta + 0 + (R_2 + L_{02} p) I_{\beta 2} = 0 \quad (22) \end{aligned}$$

where

$L_{01} = (L_1 - M_{c1})$	is the apparent self-induction of the three-phase winding of the stator of the first machine
$l_{01} = (l_1 - M_{p1})$	is the apparent self-induction of the three-phase winding of the rotor of the first machine
$M_{01} = 1{\cdot}5\, M_1$	is the apparent mutual induction of the three phases of stator and rotor
$l_{02} = (l_2 - M_{p2})$	is the apparent self-induction of the three phases of the rotor of the second machine
$L_{02} = (L_2 - M_{c2})$	is the apparent self-induction of the three phases of the stator of the second machine
$M_{02} = 1{\cdot}5\, M_2$	is the apparent mutual induction of the three phases of stator and rotor

$r_1 - r_1' = r$; $R_1' - R_p = R_2$

$l_{01} - l_{02} = l_0$

For many operating conditions of the set it can be assumed that the rotor speed $p\Theta$ is constant. The correctness of this assumption is obvious in respect to (1) the stationary operating range, and (2) transient changes in the magnetic fields during the occurrence of slight changes in the rotor speed.

COMPUTATION OF PULLING FORCE AND STRESS DISTRIBUTION IN THE DRAWING OF WIRES.

By T. POESCHL. (From *Metallwirtschaft*, Vol. 23, 1944, pp. 245-249.)

THE development of a mathematical theory of the wire drawing process was undertaken in order to gain better insight into the magnitude and the distribution of stresses and deformations obtaining in the die and to establish the relationship of these factors with the pulling force and the drawing speed.

Upon introduction of the cylindrical wire into the die in stress-free state the wire is subjected to a tri-axial state of stress which soon exceeds the plastic limit as the wire passes into the die. It will be assumed that at every point of the plastic region the components of the stresses, or at least some of them, have attained certain

limit values beyond which the elastic behaviour ceases and the plastic flow commences. The plastic limit therefore is represented by a tensor which is termed the yield point tensor. This is an adaptation of the usual yield point concept to the state of tri-axial stress. It cannot, however, be assumed that the yield point definition for conditions of tri-axial stress will provide information regarding the behaviour of the material under conditions of plastic deformation once the yield point has been exceeded.

If the material were to retain its purely elastic properties when passing through the die, the component stresses σ_r, $\sigma_\vartheta = \sigma_\varphi$, and $\tau_{r\vartheta}$ would have to increase continually from the inlet to the outlet of the die. But if the material assumes plastic state—and this will in most cases take place very soon upon entrance of the material into the die—it is but reasonable to assume that beyond the plastic limit the stresses continue to increase in the same direction and will attain their maximum near the outlet of the die. That this is actually the case is proven by the progressive smoothening of the bore of the die and its progressive wear towards the outlet. Any decrease in the stresses towards the outlet of the die is therefore out of the question. The increase in the component stresses as the wire progresses through the die is reflected by an increase in the tensile strength of the material.

The law governing the increase in strength may be expressed in its simplest form by assuming the existence of a linear relationship between the stress and the plastic deformation. This relationship resembles that existing in the elastic region, with Young's modulus being replaced by an increase-in-strength factor V. The latter factor is numerically smaller than Young's modulus of elasticity E and is assumed to remain constant. If the displacements are replaced by velocities of displacement, then the factor V will have to be expressed in kgsec. per sq. cm. instead of in kg. per sq. cm. The reciprocal ν of Poisson's ratio is also to be replaced by a coefficient of transversal plasticity ν' which may be taken as $\nu' = 0{\cdot}48$ for the purpose of this investigation.

During the plastic deformation in the wire drawing process a small but finite change in volume is taking place. Actually this change in volume amounts to only a very small percentage, but the presence of this volume change is essential to the solubility of the linear stress-deformation equations applying to the plastic range and with it to the establishment of a direct relationship between stress and deformation.

The introduction of the yield point tensor implies the existence of an initial state of stress in the die, the yield point tensor being given by

$$\begin{bmatrix} 0, & \overset{(s)}{\tau_{r\vartheta}}, & 0 \\ \overset{(s)}{\tau_{r\vartheta}}, & \overset{(s)}{\sigma_\vartheta}, & 0 \\ 0, & 0, & \overset{(s)}{\sigma\varphi} \end{bmatrix}$$

where it will be assumed that $\overset{(s)}{\sigma_\vartheta} = \overset{(s)}{\sigma_\varphi}$.

This concept rests upon the following assumptions, to wit :—

(*a*) The normal stress prevailing at the entrance to the die is taken to be zero, that is, the work piece is introduced into the die in stress-free state. The radial component of the yield point tensor will also be assumed either to equal zero or to be at least very small, although the transversal stress and the circumferential stress at the die bore have already risen to the yield point $\overset{(s)}{\sigma_\vartheta}$.

(*b*) The transversal stress and the peripheral stress at the inlet of the die have an initial value of $\overset{(s)}{\sigma_\vartheta} = \overset{(s)}{\sigma\varphi}$. For the sake of simplification this value will be assumed to coincide with the yield point of the material.

(*c*) The shear stress at the die inlet takes its inception from an initial limit value $\overset{(s)}{r_{r\vartheta}}$. Where it is considered appropriate to introduce a friction factor (f), it will be taken $\overset{(s)}{r_{r\vartheta}} = f\,\overset{(s)}{\sigma_\vartheta}$ and f will be considered to have a constant value.

If the yield point tensor given above is taken to represent the initial state of equilibrium, linear equations can be established for the components of that tensor, these components applying to the plastic state. Also all components will correspond to an identical value of the increase-in-strength factor V. In this case the actual stresses will be given by

$$\sigma_r = \frac{V}{1+\nu'}\left[\epsilon_r + \frac{\nu'}{1-2\nu'}(\epsilon_r + 2\,\epsilon_\vartheta)\right] \quad .. \quad (1)$$

$$\sigma_\vartheta = \sigma_\varphi = \frac{V}{1+\nu'}\left[\epsilon_\vartheta + \frac{\nu'}{1-2\nu'}(\epsilon_r + 2\,\epsilon_\vartheta)\right] \quad (2)$$

$$\tau_{r\vartheta} = \frac{V}{2(1+\nu')}\gamma_{r\vartheta} \quad .. \quad .. \quad (3)$$

The components of the displacement tensor are determined solely by the radial displacement $u = u(r, \vartheta)$ and it thus is :

$$\epsilon_r = \frac{\partial u}{\partial r}\,;\quad \epsilon_\vartheta = \epsilon\varphi = \frac{U}{r}\,;\quad \gamma_{r\vartheta} = \frac{1}{r}\,\frac{\partial u}{\partial \vartheta} \quad .. \quad (4)$$

For a value $\nu = 0{\cdot}48$ the equations (1), (2), and (3) can therefore be written

$$\sigma_\nu = \frac{V}{1{\cdot}48}\left[\frac{\partial u}{\partial r} + 12\left(\frac{\partial u}{\partial r} + \frac{2u}{r}\right)\right] = \frac{V}{1{\cdot}48}\left[13\,\frac{\partial u}{\partial r} + 24\,\frac{u}{r}\right] \quad (5)$$

$$\sigma_\vartheta = \sigma_\varphi = \frac{V}{1{\cdot}48}\left[\frac{u}{r} + 12\left(\frac{\partial u}{\partial r} + \frac{2u}{r}\right)\right] = \frac{V}{1{\cdot}48}\left[12\,\frac{\partial u}{\partial r} + 25\,\frac{u}{r}\right] \quad (6)$$

$$\tau_{r\vartheta} = \frac{V}{1{\cdot}48}\,\frac{1}{2v}\,\frac{\partial u}{\partial \vartheta} \quad .. \quad .. \quad (7)$$

The state of stress corresponding to the aforementioned yield point tensor according to definition represents a state of equilibrium, and it will not therefore be considered to exert any influence upon the stresses occurring within the range of strength increase. Consequently, the latter stresses can be expressed by the same equations as established above for the elastic range. Only the " elastic " constant and the boundary conditions will be different. By integration, the following solution obtains, to wit :—

$$u = u(r, \vartheta) = a\,[1 - (1-\nu')\,(5-4\nu')\,\vartheta^2]\,r^{-4(1-\nu')} + B_r \quad (8)$$

so that with $\nu' = 0{\cdot}48$, it becomes

$$u = u(r, \vartheta) = a\,(1 - 1{\cdot}6\,\vartheta^2)\,r^{-2{\cdot}08} + Br. \quad .. \quad (9)$$

The integration constants a and B can be obtained from the following two boundary conditions :

(*a*) The total stress will be assumed to be zero at the spherical entrance section, the shear stress components in the direction of the axis of the die being neglected, so that for $r = r_1$, it becomes

$$Z_1 = 2\pi\,r_1^2 \int_0^a \sigma_r\,(r_1, \vartheta)\,d\vartheta = 0 \quad .. \quad (10)$$

This means that no force is exerted upon the material as it enters the die.

(*b*) At the inlet edge of the die the normal stress, normal that is relative to the die, will be assumed to equal the yield point in compression. For $r = r_1$, it will then be

$$\vartheta = a : \sigma_\vartheta (r_1, a) = \sigma_\varphi (r_1 a) = \overset{(s)}{\sigma_\vartheta} \quad .. \qquad .. \qquad (11)$$

These two boundary conditions determine the integration constants a and B in equations (8) and (9). The solution is therefore complete and all other factors of the drawing process, such as stress components, drawing pull, velocity of the wire entering and leaving the die, and the components ϵ_r, $\epsilon_\vartheta = \epsilon_\varphi$ and $\gamma_{r\vartheta}$ are therefore determined.

The integration constants can be computed by introducing the solution given by formula (9) into the boundary condition equations (10) and (11). Assuming a value $\nu' = 0{\cdot}48$, we obtain

$$a = \frac{r_1^{-3{\cdot}08}}{3{\cdot}04\,(1-0{\cdot}8a^2) + 0{\cdot}04\,(1-1{\cdot}6a^2)} \cdot \frac{1{\cdot}48\,\overset{(s)}{\sigma_\vartheta}}{V} \qquad (12)$$

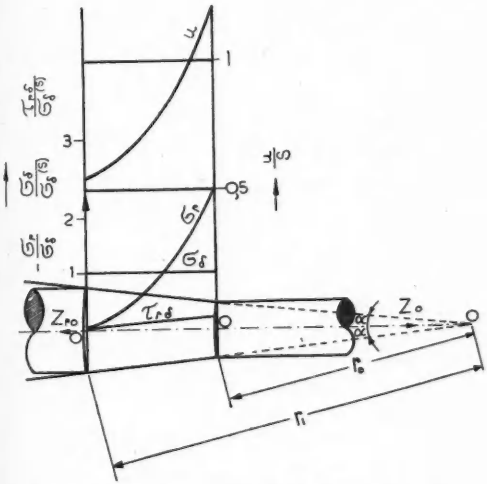


Fig. 1. **Diagram showing stress distribution in wire passing through the die.**

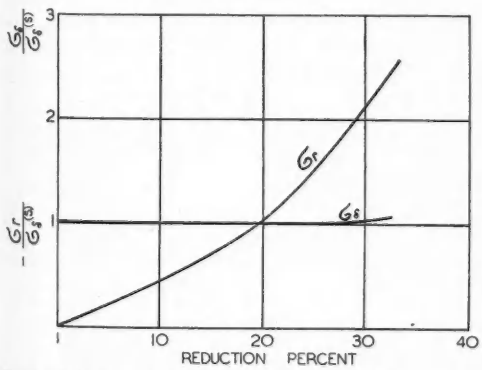


Fig. 2. **Influence of the reduction upon the wire stresses.**

$$B = \frac{3{\cdot}04\,(1-0{\cdot}8a^2)}{3{\cdot}04\,(1-0{\cdot}8a^2) + 0{\cdot}04\,(1-1{\cdot}6a^2)} \cdot \frac{1{\cdot}48\,\overset{(s)}{\sigma_\vartheta}}{37\,V} \qquad (13)$$

The conditions of stress and deformation prevailing at every point (r, ϑ) of the die are thus determined by the following expressions :

$$\left.\begin{aligned}
u &= a\,(1-1{\cdot}6\,\vartheta^2)\,r^{-2{\cdot}08} + B_r \\
\epsilon_r &= \frac{\partial u}{\partial_r} = -\,2{\cdot}08\,a\,(1-1{\cdot}6\,\vartheta^2)\,r^{-3{\cdot}08} + B \\
\epsilon_\vartheta &= \frac{u}{r} = a\,(1-1{\cdot}6\,\vartheta^2)\,r^{-3{\cdot}08} + B \\
\gamma_{r\vartheta} &= \frac{1}{r}\frac{\partial u}{\partial \vartheta} = -\,3{\cdot}2\,a\,r^{-2{\cdot}08}\,\vartheta \\
\sigma_r &= \frac{V}{1{\cdot}48}\,(13\,\epsilon_r + 24\,\epsilon_\vartheta) \\
\sigma_\vartheta &= \frac{V}{1{\cdot}48}\,(12\,\epsilon_r + 25\,\epsilon_\vartheta) \\
\tau_{r\vartheta} &= -\,\frac{V}{1{\cdot}48}\,1{\cdot}6\,a\,r^{-2{\cdot}08}\,\vartheta
\end{aligned}\right\} \qquad (14)$$

Finally the volume change is given by

$$\Theta = \epsilon_r' + 2\epsilon_\vartheta = \frac{1-2\nu'}{V}\,(\sigma_r + 2\sigma_\vartheta) = \frac{0{\cdot}04}{V}\,(\sigma_r + 2\sigma_\vartheta) \qquad (15)$$

Assuming a tapering die with $r_1 = 1{\cdot}5$ cm., $r_0 = 1{\cdot}0$ cm., and $a = 0{\cdot}1$ (corresponding to an angle of 5·7 deg.) the inlet and outlet diameters of the die are $2R_1 = 0{\cdot}3$ cm. and $2R_0 = 0{\cdot}2$ cm., and the linear reduction is therefore 33 per cent.

With $\nu' = 0{\cdot}48$ and $1{\cdot}48\ \overset{(s)}{\sigma_\vartheta} = S$ .. (16)

we obtain $a = 1{\cdot}142\ S$ and $B = 0{\cdot}0267\ S$

The equations (14) and (15) thus yield the following solutions :

Entering the die	*Leaving the die*
$u\,(r_1,0) = 0{\cdot}532S$	$U_o = u\,(r_o,0) = 1{\cdot}169S$
$\epsilon_r\,(r_1,0) = -\,0{\cdot}654S$	$\epsilon_r\,(r_o,0) = -\,2{\cdot}35S$
$\epsilon_\vartheta\,(r_1,0) = 0{\cdot}355S$	$\epsilon_\vartheta\,(r_o,0) = 1{\cdot}169S$
$\Theta\,(r_1,0) = 0{\cdot}056S$	$\Theta\,(r_o,0) = -\,0{\cdot}01S$
$\sigma_r(r_1,0) = 0$ (as assumed)	$\sigma_r(r_o,0) = -\,2{\cdot}47\,\overset{(s)}{\sigma_\vartheta}$
$\sigma_\vartheta(r_1,0) = \overset{(s)}{\sigma_\vartheta}$,,	$\sigma_\vartheta(r_o,0) = 1{\cdot}05\,\overset{(s)}{\sigma_\vartheta}$
$\tau_{r\vartheta}\,(r_1,a) = -\,0{\cdot}7087\,\overset{(s)}{\sigma_\vartheta}$	$\tau_{r,\vartheta}(r_o,a) = -\,0{\cdot}183\,\overset{(s)}{\sigma_\vartheta}$

It should be noted that the longitudinal stress increases very considerably while the transversal stress does not diminish as would be expected on the basis of the popular hypothesis of plasticity, but actually increases slightly. With the assumed reduction of 33 per cent the longitudinal stress would increase to a value 2·47 times that of the yield point in compression. This stress would appear to be in excess of the carrying capacity of the wire. It must also be pointed out that the volume change at the die inlet has the negative sign, which is due to the boundary conditions assumed. At the outlet it is, however, positive, that is to say, an elastic material would exhibit a small permanent volume increase when passing through the die. However, the influence of the transition from the elastic to the plastic condition upon the volume change still remains to be clarified.

The equations (14) and (15) may be used to chart the stress distribution and the volume change of the wire as it passes through the die. (Fig. 1). If the reduction is assumed as 10 per cent instead of 33 per cent, the longitudinal stress is found to be no more than about 46 per cent of the yield point in compression. The transversal stress changes but little throughout the die and it can therefore be considered as constant. The relation between the longitudinal stress σ_r and the transversal and peripheral stresses $\sigma_\vartheta = \sigma_\varphi$ as function of the reduction is given in Fig. 2.

TEMPERATURE DISTRIBUTION AND HEAT FLOW IN THE EXHAUST VALVES OF DIESEL ENGINES

By I. DROLSHAMMER. (From *Mitteilungen aus dem Inst. Thermodyn. & Verbrenn'gsmot. T.U. Zurich*, No. 3, 1943).

TEMPERATURE measurements were made on the valves of two diesel engines. The first was a single cylinder four-stroke cycle high speed diesel specially built as an experimental engine by the Saurer Company for the Federal Technical University, Zurich ; while the second unit was an experimental three-cylinder low speed engine built by the Swiss Locomotive Works, Winterthur. The abstract given below refers solely to the latter engine.

The three-cylinder engine is designed for normal

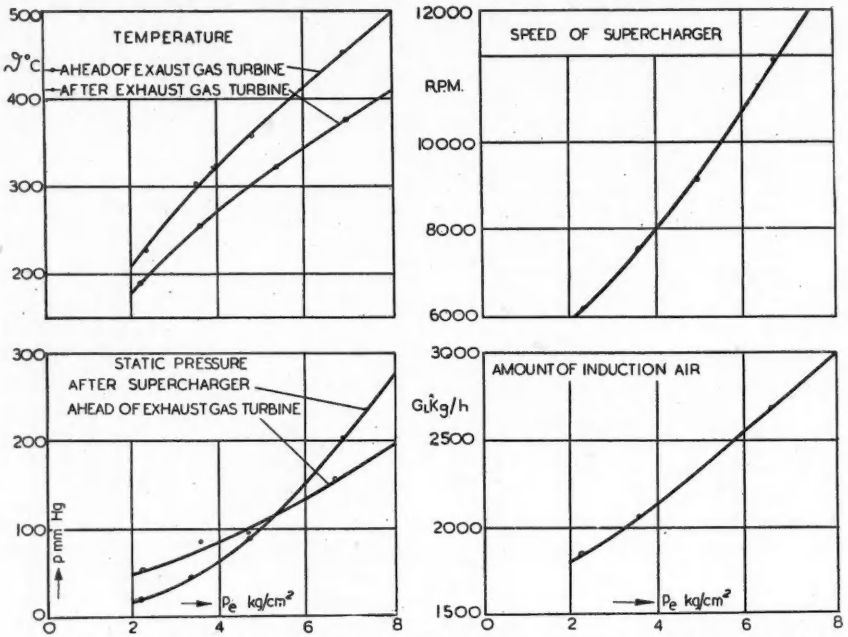


Fig. 1. Characteristics of engine when operated with super-charger at different loads with engine speed n = 300 r.p.m. = constant.

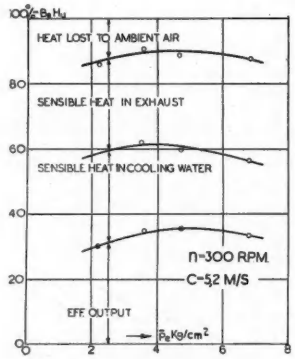


Fig. 2. Heat balance of engine when operated with supercharger.

operation with a Büchi supercharger. A few test runs were, however, also made with the supercharger removed and with the original valve cams replaced by cams more suitable for operation without supercharger. The engine data are the following :

Bore	390 mm.
Stroke	520 mm.
Connecting rod length	1250 mm.
Power developed with supercharger ..	450 h.p.
„ „ without „ ..	350 h.p.
Normal engine speed	300 r.p.m.

The engine is built for mechanical injection, the valves being operated by push rods and rocker arms. The valve timing is as follows :

	With supercharging	Without supercharging
Exhaust opens	44 deg. before B.D.C.	62 deg. before B.D.C.
„ closes	51 deg. after T.D.C.	35 deg. after T.D.C.
Air inlet opens	67 deg. before T.D.C.	25 deg. before T.D.C.
„ closes	28 deg. after B.D.C.	50 deg. after B.D.C.

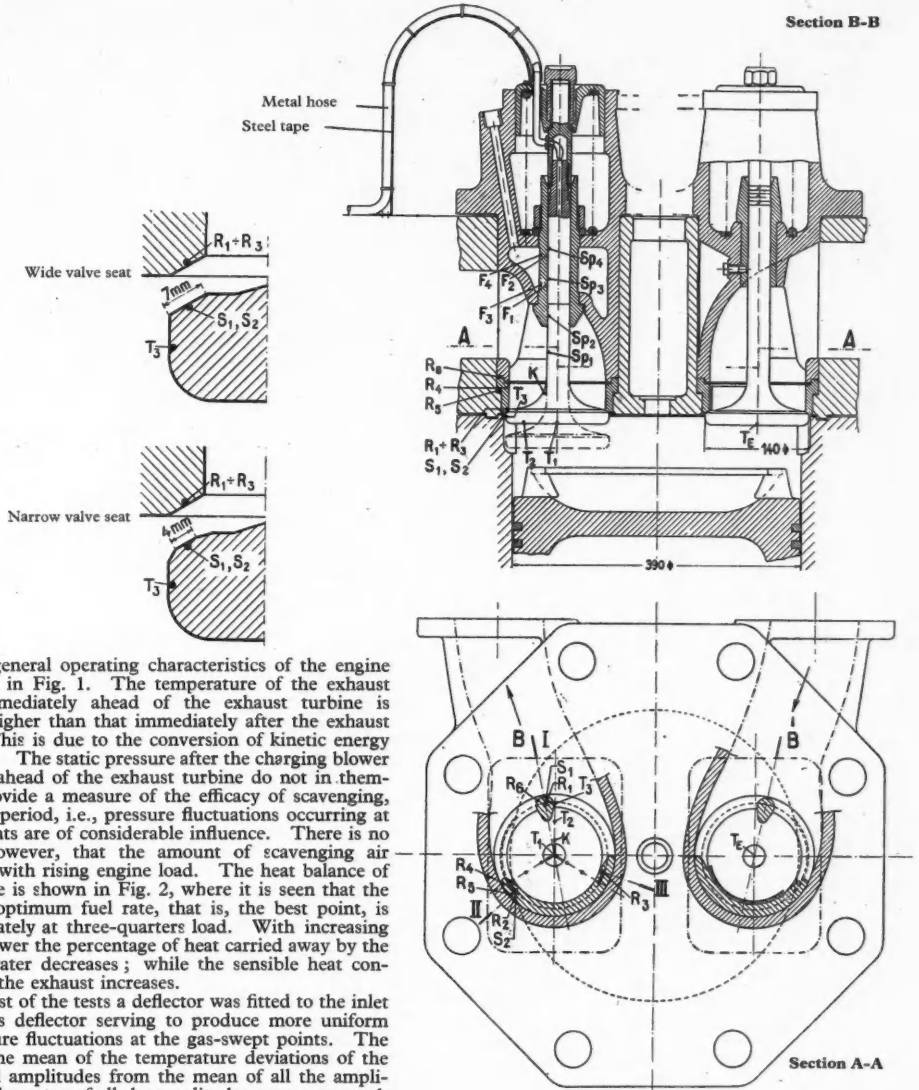


Fig. 3. **Location of thermocouples.**

The general operating characteristics of the engine are given in Fig. 1. The temperature of the exhaust gases immediately ahead of the exhaust turbine is slightly higher than that immediately after the exhaust valve. This is due to the conversion of kinetic energy into heat. The static pressure after the charging blower and that ahead of the exhaust turbine do not in themselves provide a measure of the efficacy of scavenging, since the period, i.e., pressure fluctuations occurring at these points are of considerable influence. There is no doubt, however, that the amount of scavenging air increases with rising engine load. The heat balance of the engine is shown in Fig. 2, where it is seen that the point of optimum fuel rate, that is, the best point, is approximately at three-quarters load. With increasing engine power the percentage of heat carried away by the cooling water decreases; while the sensible heat contained in the exhaust increases.

In most of the tests a deflector was fitted to the inlet valve, this deflector serving to produce more uniform temperature fluctuations at the gas-swept points. The ratio of the mean of the temperature deviations of the individual amplitudes from the mean of all the amplitudes to the mean of all the amplitudes rose to as much as 15 per cent at fractional load. This value was reduced to 5 per cent by the installation of the aforementioned deflector, which, it is assumed, produced a more uniform distribution of the fuel.

For the purpose of measuring the valve temperatures 17 thermocouples were installed; of these six were placed in the exhaust valve (Fig. 3); another six in the seat; four in the valve stem guide; and one in the inlet valve. The exhaust valve can be rotated into three positions with a mutual angularity of 120 degrees, hereby enabling observation of the temperature distribution over the valve periphery. The thermocouples in the seat are so arranged that regardless of the valve position, two thermocouples in the seat are always located opposite two thermocouples in the exhaust valve.

The valve temperatures measured are charted in Figs. 4, 5 and 6. Referring to Fig. 4 it will be seen that the influence of the engine speed upon the temperatures of the exhaust valve is not great; at the point considered the temperature is seen to drop by no more than approx. 30 degrees if the engine speed is decreased from 300 r.p.m. to 200 r.p.m.

The influence of the Büchi supercharging process is illustrated in Fig. 5, which shows the temperature variations obtaining with different mean effective pressures. The temperature of the exhaust valve was taken at point T_1 and that of the inlet valve at point T_E (Fig. 3); while the exhaust gas temperature was recorded at the outlet in the cylinder head. It will be seen that at full engine load, corresponding to a m.e.p. of

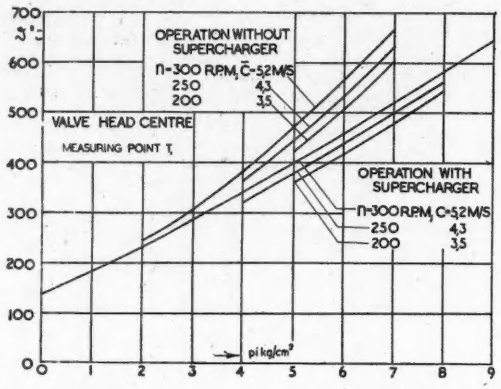


Fig. 4. **Exhaust valve temperature *versus* engine load and speed for operation with and without supercharger.**

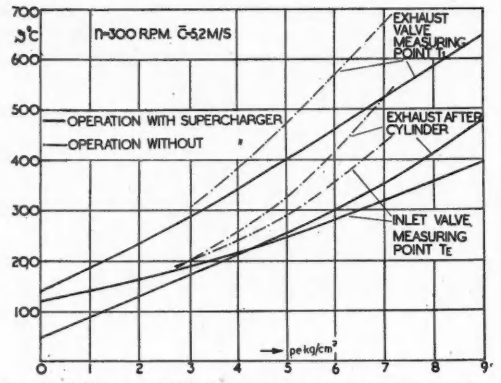


Fig. 5. **Influence of Büchi supercharging upon valve and exhaust temperatures.**

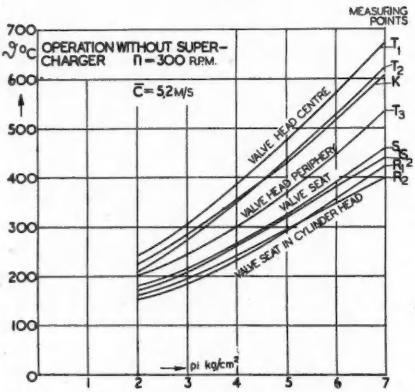


Fig. 6. **Exhaust valve temperature variation with engine load for operation without supercharger.**

9 kg. per sq. cm. with supercharging, and of 7 kg. per sq. cm. without supercharging, the respective temperatures of the exhaust valve are approx. 220 deg. C. below those of the inlet valve. At full load both with and without supercharging, the temperature difference

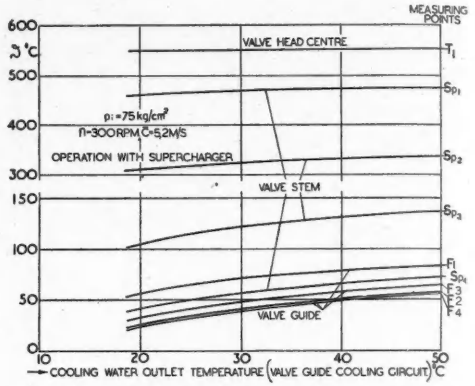


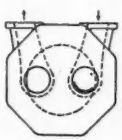
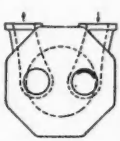
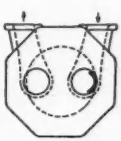
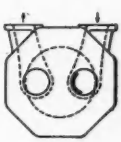
Fig. 7. **Influence of valve guide cooling upon exhaust valve temperature.**

between the hottest and the coldest points of the exhaust valve amounted to some 150–160 deg. C. At the root of the valve (measuring point *K*) the temperature was always about 90–120 deg. C. above that of the exhaust gases, the valve yielding heat to the exhaust gases. Fig. 5 indicates that the employment of supercharging considerably reduces exhaust valve temperatures. In the case of point T_1 for instance, supercharging is seen to reduce the temperature by some 140 deg. C., the load corresponding to a mean indicated pressure of 7 kg. per sq. cm. ; while the corresponding decrease in the temperature of the inlet valve amounts to 100 deg. C. This may be partly attributed to the larger amount of excess air supplied to the cylinder when resorting to pressure charging. Also, because of the considerable overlap of exhaust and inlet valve opening near the T.D.C. position, a scavenging effect is obtained. At full load the amount of air taken up by the scavenging process amounts to some 27 per cent. of the total air supply. With identical valve temperatures the employment of Büchi-charging secures an increase in effective power by 28 per cent.

In order to ascertain the influence of the width of the valve seat upon the valve temperature, the seat width was reduced from 7 mm. to 4 mm. and the test runs repeated under the same conditions as before. It was found that the exhaust valve temperatures were not materially changed. At measuring point T_1 the temperature remained the same as before, as was also the temperature at point *K*. A temperature increase, however, obtained towards the periphery of the valve head, the temperature at point S_1 having increased by some 40 deg. C., and at point R_1 by 10 deg. C. The permanent temperature difference between S_1 (valve) and R_1 (valve seat) was therefore 30 deg. C. higher in the case of the narrow valve seat. At point S_2, the temperature increased by 12 deg. C., but at the juxtaposed point R_2 it fell by 15 deg. C. causing an increase in temperature difference by 27 deg. C., that is to say, a temperature difference of 67 deg. C.

The respective influences of the cooling of the cylinder and of the valve guide upon the exhaust valve temperatures were investigated separately. The temperature of the water leaving the cooling circuit of the valve guide was kept at 20 deg. C., and the water exit temperature of the cylinder cooling circuit was then reduced from 65 deg. C. to 35 deg. C. by increasing the flow rate with an inlet temperature of 10 deg. C. This resulted in a decrease of the valve temperature at point T_1 from 564 deg. C. to 562 deg. C. ; while at point S_2 a temperature decrease from 367·5 deg. C. to 363 deg. C. took place. The corresponding temperature at point R_2 diminished from 350 deg. C. to 341 deg. C. The

TABLE I.

n = 300 r.p.m.		Operation without supercharger			Excess air factor = 1·98
		Air swirl			
		with		without	
Position of Deflector		 I (Normal)	 II	 III	 IV
Valve head centre	T_1	546	556	567	594
Valve head	$T_{2\,III}$	588	569	535	560
Valve head periphery	$T_{3\,III}$	538	522	485	507
Valve seat	K_{III}	517	525	496	514
Valve seat in valve head	$S_{1\,III}$	475	468	453	486
Valve seat in valve head	$S_{2\,III}$	358	489	439	467
Valve seat in cylinder head	R_1	338	343	380	393
Inlet valve head centre	T_E	329	329	365	372

influence of the exit temperature of the water from the cooling circuit of the valve stem guide is charted in Fig. 7, which is based on a water temperature of 50 deg. C. at the outlet of the cylinder cooling circuit. It will be seen from this graph that the influence of the cooling water outlet temperatures of the valve-guide cooling circuit extends solely to the metal temperatures at point S_{p3} and S_{p4}; while the temperature at the centre of the valve head (point T_1) remains unaffected. This cooling circuit therefore serves exclusively for safeguarding trouble-free running of the valve stem in its guide.

In order to ascertain the influence of the deflector with which the inlet valve is equipped, four different test runs were made; the valve, and with it the deflector, being rotated into the different positions indicated at the top of Table I. The temperatures given in this table are based upon an identical excess air factor of 1·98, the engine operating without the Büchi charger. According to the position of the deflector, the amount of induction air varies by 3 to 4 per cent; while it increases by some 12 per cent if the deflector is removed.

If the deflector is placed in the position I and II indicated in Table I., a vortex is produced in the combustion chamber, a tangential velocity component being imparted to the induction air as it enters the combustion chamber. When located in position III, the deflector directs the air towards the exhaust valve; and with the deflector in position IV the air is directed towards the cylinder wall. The temperatures of the exhaust valve are notably influenced by the direction of the air current thus produced. For position IV, the centre of the valve head attains maximum temperature at point T_1. In position III the exhaust valve is especially well cooled where it faces the inlet valve (points T_{2m}, T_{3m}); while these points are hottest when the deflector is in its normal position I. The temperature of the inlet valve itself is lowest when tangential motion is imparted to the induction air.

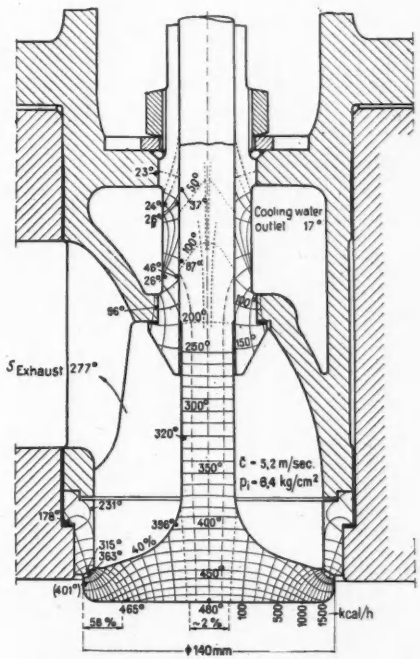


Fig. 8. **Temperature distribution in exhaust valve n = 300 r.p.m., m.i.p. = 6·4 kg. per sq. cm. Inlet valve not masked.**

STATIONARY HEAT FLOW IN THE EXHAUST VALVE.

The temperature distributions in the valve body and seat, as shown in Figs. 8 and 9, were obtained by analytical evaluation of the temperatures measured. But in the view of the temperature variations over the circumference of the valve, these graphs can no more than approximately indicate the flow of heat in the valve body. The chief conclusions to be drawn from Figs. 8 and 9 are as follows :

About two-thirds to three-fourths of the heat absorbed by the valve from the products of combustion are rejected through the valve seat. The head of the valve and the root of the valve stem yield from one-third to one-quarter of the absorbed heat to the spent gases ; while the amount of heat dispersed by the valve stem is insignificant. The root of the valve stem does not receive any heat from the exhaust gases. There is thus no need for improving the heat flow through the valve stem by making it of the hollow sodium filled type. Sufficient metal area must, however, be provided for conduction of the heat to the valve seat, so as to avoid the occurrence of large temperature differences between centre of the valve head and the valve seat.

In Fig. 9 are shown the respective temperature distributions obtaining with and without deflector, the engine operating supercharged and at full load. The topmost part of the engraving shows the temperature distribution obtaining with the valve seat 7 mm. wide ; while the temperatures measured with the valve seat

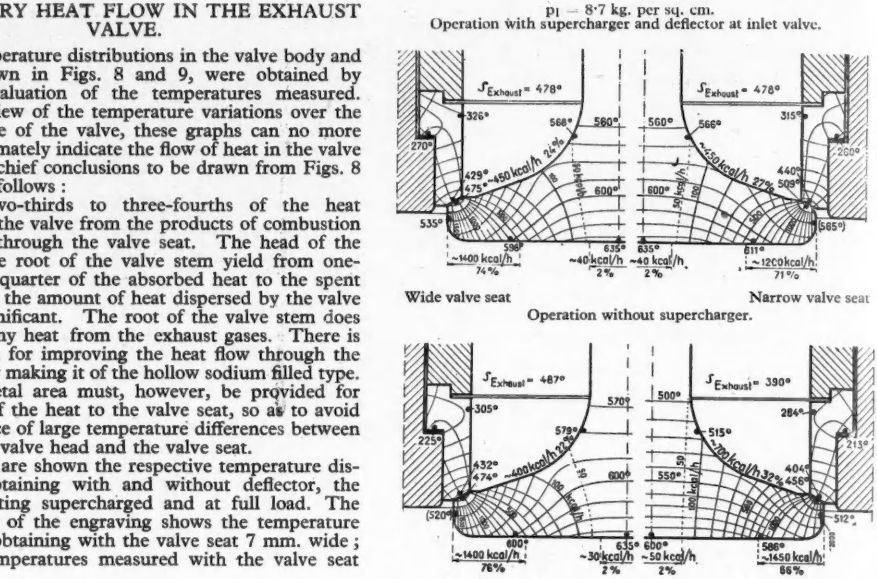

p1 = 8·7 kg. per sq. cm.
Operation with supercharger and deflector at inlet valve.
Wide valve seat
Narrow valve seat
Operation without supercharger.
p1 = 6·7 kg. per sq. cm.
Inlet valve with deflector
p1 = 6·6 kg. per sq. cm.
Inlet valve without deflector

Fig. 9. **Influence of supercharging, width of valve seating, and valve masking upon temperature distribution in exhaust valve n=300 r.p.m.**

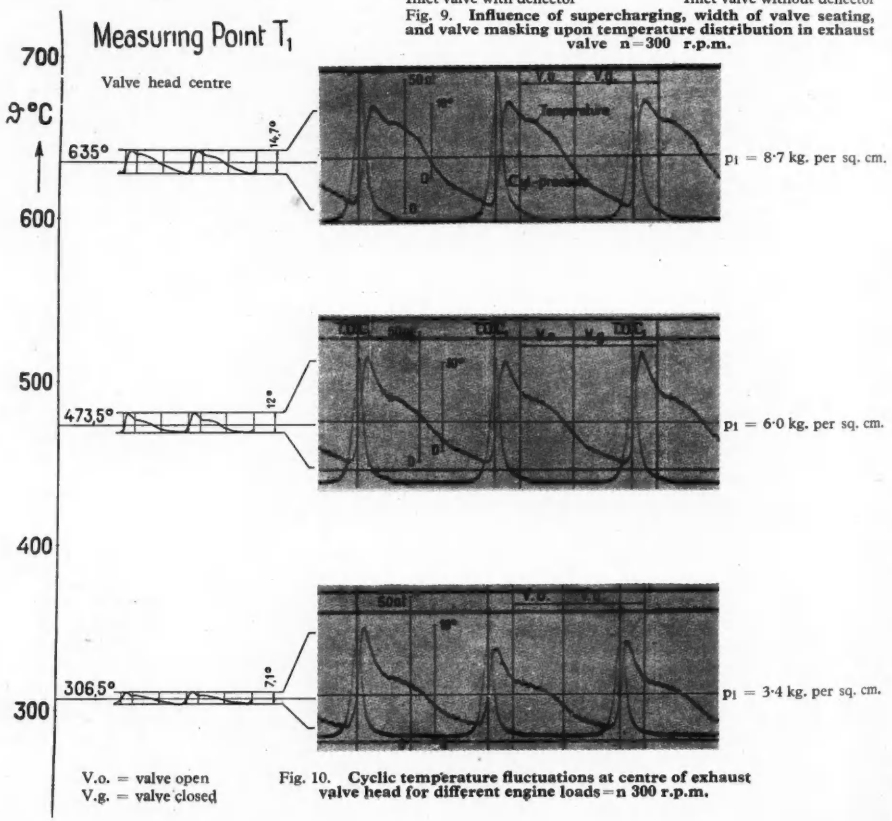


Fig. 10. **Cyclic temperature fluctuations at centre of exhaust valve head for different engine loads=n 300 r.p.m.**

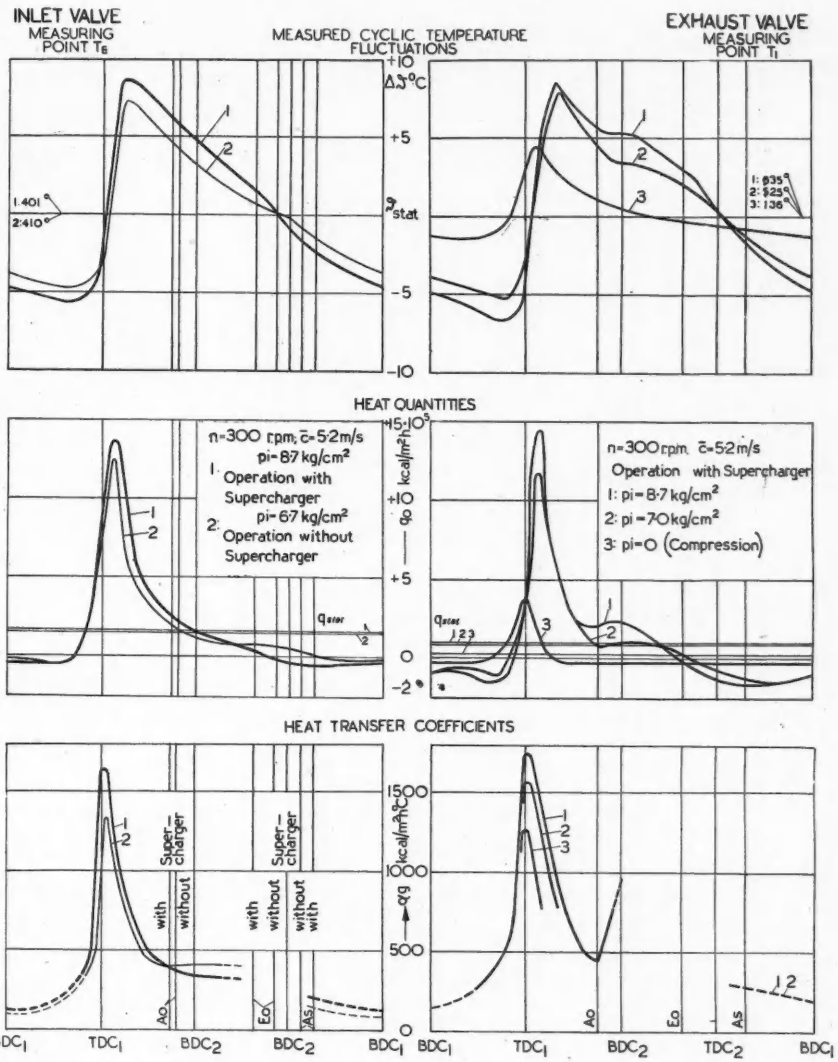


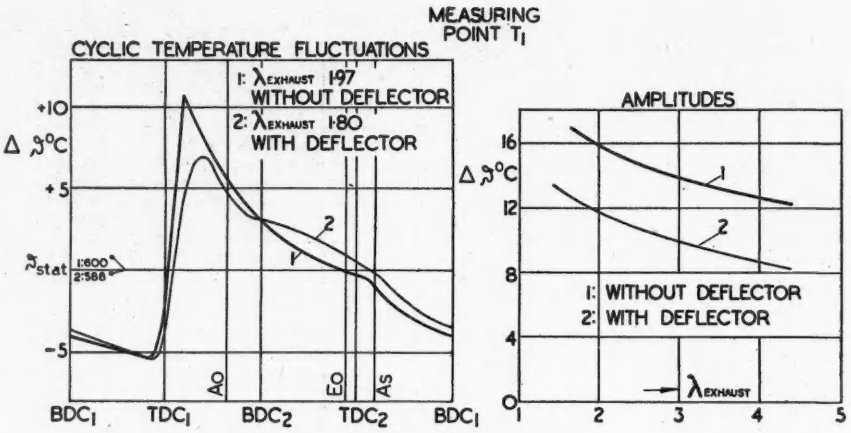
Fig. 11. **Evaluation of valve temperature diagrams.**

reduced to 4 mm. width are charted in the lower graph. Of the heat absorbed by the valve, 74 per cent is removed through the 7 mm. valve seat and 71 per cent through the 4 mm. valve seat, that is to say, the specific heat flux by conduction through the 4 mm. valve seat is 1·5 times that passing through the 7 mm. seat. With a seat width of 7 mm. the back of the valve head and the root of the valve stem yields 24 per cent of the absorbed heat to the exhaust gas, and 27 per cent with the 4 mm. valve seat. Two per cent of the heat absorbed by the valve is removed through the valve stem.

The amount of heat given off to the exhaust gas increases if the deflector is removed. Thus when the engine is run without the supercharger and without the valve deflector, the valve yields 32 per cent of its absorbed heat to the exhaust gases; and this figure increases to 40 per cent when supercharging (see Fig. 9, lower graph r.h. side).

CYCLIC VARIATIONS IN TEMPERATURE AND HEAT FLUX.

Fig. 10 shows the cyclic temperature fluctuations at point T_1 for three different engine loads at identical engine speed. The indicated pressure diagrams obtained with a piezo-electric indicator are also shown. It was found that the temperature at points T_1, T_2, T_3, and T_E rose sharply during the combustion process, while the subsequent drop in temperature at point T_1, T_2, and T_3 slows down abruptly when the exhaust valve begins to open; and at the points T_{3I}, T_{3II}, and T_{3III} of the periphery of the valve head (Fig. 3) the metal temperature even shows a temporary rise. That this rise is due to the high gas velocity and to the consequently high heat transfer rate prevailing at this point, is seen from the fact that no such temperature rise takes place at point T_E of the inlet valve, which is not swept



Figs. 12a, 12b. Influence of inlet valve masking upon cyclic temperature fluctuations at centre of exhaust valve head.

by the stream of the exhaust gases. The temporary temperature rise as such is largely due to the masking of the inlet valve, the swirl created by the deflector still prevailing in the exhaust period. This effect is illustrated by Fig. 12a, which shows the cyclic temperature fluctuations obtaining at point T_1 with and without deflector respectively. As will be seen from Fig. 12b, the presence of the deflector also serves to depress the amplitude of the cyclic temperature variation at point T_1 by some 25 per cent.

The graphs given in Fig. 11 show the results of an evaluation of the temperature measurements taken at points T_E (centre of inlet valve head) and T_1 (centre of exhaust valve head). The temperatures recorded at these points are given in the upper two graphs of Fig. 11. These temperature data were used to compute the amount of heat transferred to the valve at every instant of the cycle, thus obtaining the two charts given in the centre of Fig. 11. The coefficients of heat tranfer obtaining between the gas contents of the combustion chamber and the valve head can then be found as

$$\alpha_g = \frac{q}{\vartheta_g - \vartheta_v},$$

where q is the amount of heat transferred in kal per sq. m. per hour, ϑ_g is the gas temperature and ϑ_v is the valve temperature. The gas temperature is computed from the (measured) pressure diagram with the use of an entropy diagram. The heat transfer coefficients computed in this way are plotted in the charts shown at the bottom of Fig. 11.

PERMEABILITY OF CONCRETE.

By L. Boyd Mercer. (From *The Commonwealth Engineer*, July 2nd and August 1st, 1945, Vol. 32, No. 12, pp. 349-357, and Vol. 33, No. 1, pp. 12-17.)

Permeability of concrete may be defined as that property of concrete which permits the passage of a liquid, usually water, into and through the body of the concrete when subjected to pressure. Such passage is usually considered to occur without any impairment of the structure or displacement of parts, but leaching or corrosion is a natural corollary. The importance of securing impermeable concrete has long been realised, but the means by which this desirable property may be obtained have been rather obscure. Testing of concrete for water retaining structures has frequently placed some reliance upon some undefined connection between permeability and the compressive strength.

The formal definition of the permeability of a porous medium may be stated as the volume of fluid of unit viscosity passing through a unit cross section of the medium in unit time under the action of a unit pressure gradient. It is thus determined entirely by the actual pore openings and general structure of the medium and is independent of the fluid.

Permeability is quite distinct from porosity which is only concerned with the relationship of the volume of pores to total volume. It is conceivable, though improbable, that a concrete could be porous, but impermeable if it contained a series of disconnected air pockets separated by impermeable material.

Insomuch as the water passages through the body of the concrete are small, it is generally accepted that D'Arcy's law for viscous flow applies. This enables us to express the " Coefficient of Permeability " as flow in cubic feet per second through an area of one square foot under an hydraulic gradient of one foot per foot, thus :

$$K = \frac{Q.L.}{A.H.}$$

Where K = coefficient of permeability.
Q = flow in cubic ft. per second.
L = thickness of the medium in ft.
A = area subjected to flow, in sq. ft.
H = total difference of head, in ft.

An indication of the factors upon which K depends can be obtained from the equation for flow in capillary tubes, known as Poiseuille's law :

$$v_1 = \frac{\rho \cdot g \cdot i_1 r^2}{8\mu}$$

Where v_1 = mean velocity of flow in tube,
ρ = density of fluid, water at temperature under consideration,
g = acceleration due to gravity,
$i_1 = h/l_1$ = head lost per unit length of capillary, or slope of hydraulic gradient,
r = mean radius of tube,
μ = coefficient of viscosity of fluid, water, at temperature under consideration.

Now if Q is the total quantity of fluid passing through

Fig. 1. **Test cylinder section sealed in brass ring.**

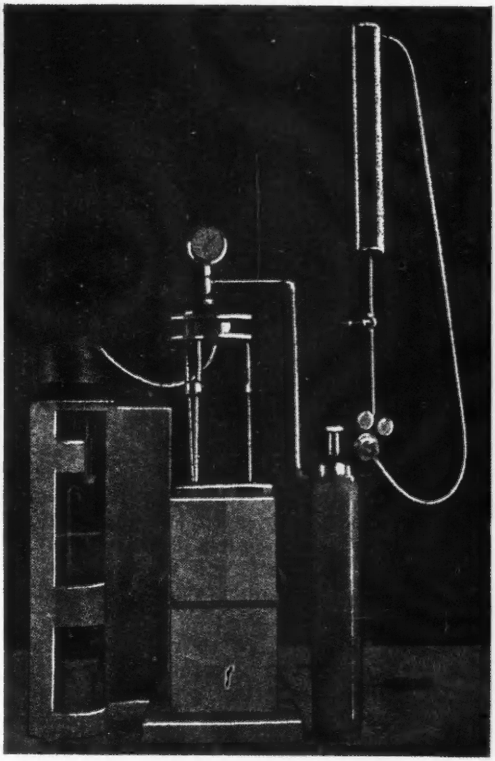


Fig. 2. **Permeability test apparatus.**

area A per second, then—

$$Q = \Sigma \frac{\pi \rho g i_1 r^4}{8\mu} = \frac{\rho g}{8\mu} \Sigma \pi r^2 \frac{h}{l_1} r^2$$

But $Q = \frac{K.A.H.}{L}$, so that for H = h,

$$K = \frac{\rho g}{8\mu} \frac{\Sigma \pi r^2}{A} \frac{L}{l_1} r^2$$

Thus it is seen that the permeability coefficient, K, can be expected to vary with the temperature, viscosity, and density of the fluid; with $\Sigma \pi r^2/A$, a function of the

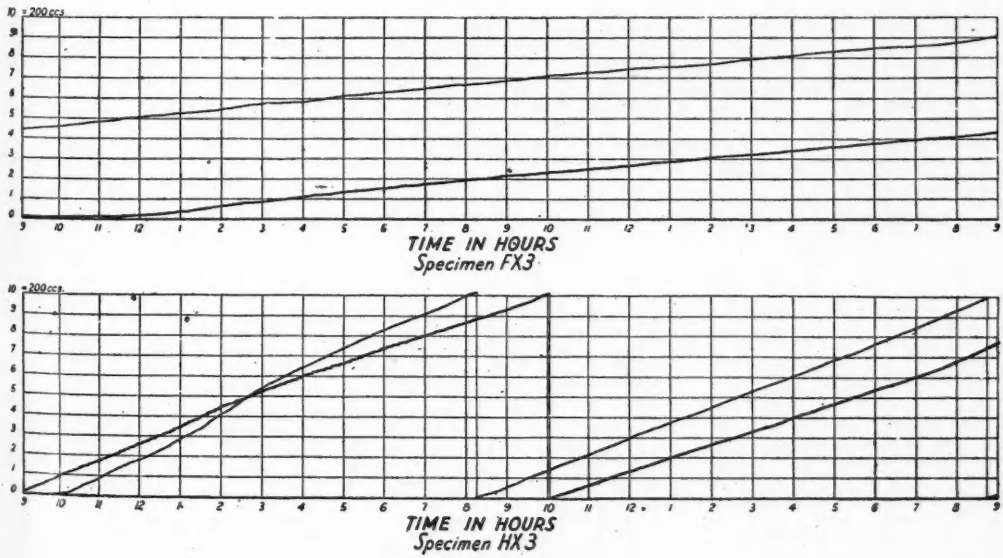


Fig. 3. **Pluviograph charts.**

porosity of the specimen ; with L/l_1 the ratio of apparent to true length of capillary tube ; and with r^2, the area of the pore.

Several methods of test for the permeability of concrete have been developed and used in research programmes of various magnitudes, but no method has become generally accepted as standard procedure.

The apparatus developed and used in research investigations at the University of Tasmania aimed at combining the attractive features from other known equipment. The desire to produce rapid results led to the use of a comparatively high pressure, 100 lb. per sq. in., and perhaps unduly thin discs, 1½ in. thick. It is not contended that this apparatus presents a scientifically accurate evaluation of the permeability coefficient, but rather that it is a simple piece of equipment which is useful in presenting rapid and comparative results. Briefly, the procedure adopted in the author's work and equipment is as follows :—A concrete disc 1½ in. thick is cut from a 12 in. × 6 in. dia. test cylinder by two cuts to minimise end effect, placed centrally in a ⅜ in. thick brass ring with internal diameter 6⅜ in. and height 1⅝ in., and the annular space between the sides filled with hot bitumen (see Fig. 1). The brass ring with disc is then set on a rubber ring on the lower of two 10 in. diameter by ⅝ in. thick steel plates, another rubber ring and similar steel plate placed on top, and the whole clamped together by eight ½-in. bolts. Water pressure is then applied to the top surface of disc. Pressure from the mains was used in preliminary experiments, but was superseded by compressed gas in an oxygen cylinder connected to water in reservoir for purposes of obtaining positive and constant pressure. Water passing through the disc discharged into a pluviograph which recorded the rate of flow. The apparatus is shown in Fig. 2, and typical pluviograph charts in Fig. 3. Test results were recorded on a standardised form.

Perhaps the most significant conclusion to be drawn from concrete permeability tests is the necessity for avoiding hasty conclusions. Most investigations have been careful to preface their summary of results by stating that the conclusions submitted are tentative and only apply to the data presented. Nevertheless experiments appear to have verified that the percolation of water through concrete follows D'Arcy's law for viscous flow. Moreover, it has been established that permeability is a definite property of concrete and is susceptible of evaluation by means of the coefficient of permeability as outlined above.

THERMAL STRESSES IN A TUBE.

By E. PALMBLAD. (From *Teknisk Tidskrift*, Vol. 29, July 21st, 1945, pp. 821-824).

IT is of considerable technical importance to be able to compute the thermal stresses which occur in a tube subjected to external heating or cooling. The exact mathematical solution can be found as follows :

It is

$$\frac{\partial^2 \xi}{\partial x^2} + \frac{m-2}{2(m-1)} . D \frac{\partial \xi}{\partial r} + \frac{m}{2(m-1)} \frac{\partial}{\partial x} D\rho - \frac{m+1}{m-1} \alpha \frac{\partial t}{\partial x} = 0 \quad .. \quad (1)$$

$$\frac{\partial}{\partial r} D\rho + \frac{m-2}{2(m-1)} \frac{\partial^2 \rho}{\partial x^2} + \frac{m}{2(m-1)} \frac{\partial^2 \xi}{\partial r \partial x} - \frac{m+1}{m-1} \alpha \frac{\partial t}{\partial r} = 0 \quad .. \quad (2)$$

$$\sigma_x = \frac{2G}{m-2} \left[(m-1) \frac{\partial \xi}{\partial x} + D\rho - (m+1) \alpha t \right] \quad .. \quad (3)$$

$$\sigma_r = \frac{2G}{m-2} \left[\frac{\partial \xi}{\partial x} + (m-1) \frac{\partial \rho}{\partial r} + \frac{\rho}{r} - (m+1) \alpha t \right] \quad (4)$$

$$\sigma_p = \frac{2G}{m-2} \left[\frac{\partial \xi}{\partial x} + \frac{\partial \rho}{\partial r} + (m-1) \frac{\rho}{r} - (m+1) \alpha t \right] \quad (5)$$

$$\tau = G \left[\frac{\partial \xi}{\partial r} + \frac{\partial \rho}{\partial x} \right] \quad .. \quad .. \quad (6)$$

$$D = \left(\frac{\partial}{\partial r} + \frac{1}{r} \right) \quad .. \quad .. \quad (7)$$

where x and r are the co-ordinates in axial and radial direction respectively ; ξ and ρ are the elastic deformation in the directions x and r respectively.

m is Poisson's ratio ;

G is the modulus of shear

σ_x, σ_r, σ_p are component stresses in the directions x, r, and perpendicular with r ;

t is the temperature in deg. C. at the distance r from the centre ;

α is the coefficient of thermal expansion of the tube material ;

λ is the thermal conductivity of the tube material.

All dimensions of length may be given in cm., and all stresses in kg. per sq. cm.

The tube considered may have an external radius r_u and an internal radius r_l, the corresponding surface temperatures being t_u and t_l. The quantity of heat flowing in radial direction through a sector φ per unit length of tube (Fig. 1) will be given by

$$Q = \text{const.} = \pm r \varphi \lambda \frac{dt}{dr}$$

Here the positive sign applies to the case of radial heat flow from the outer tube surface towards the internal tube surface. In this case it is

$$\frac{dr}{r} = C_1 \, dt$$

By integration we have

$$\ln r = C_1 t + C_2 \, ; \; t = \frac{\ln r}{C_1} - \frac{C_2}{C_1}$$

where

$$C_1 = \frac{\ln r_u - \ln r_l}{t_u - t_l}$$

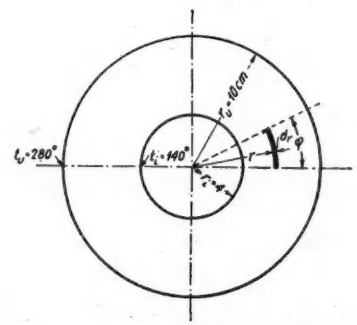


Fig. 1. Cross section through piston rod of Diesel engine.

and $$C_2 = \frac{t_u \ln r_i - t_i \ln r_u}{t_u - t_i}$$

This temperature function can also be written in the simplified form

$$t = a \ln r - b \quad .. \quad .. \quad .. \quad (8)$$

The temperature will be assumed as constant in axial direction, and it will also be assumed that the portion of the tube considered may be at sufficient distance from either free end of the tube to exclude any influence of the free ends. The thermal expansion of the tube in axial direction will then be given by

$$\xi = k \,.\, x \text{ where } k = \text{constant}$$

and it will be $$\frac{\partial \xi}{\partial x} = k \text{ and } \frac{\partial \xi}{\partial r} = 0$$

and $$\frac{\partial^2 \xi}{\partial x^2} = 0\,; \quad \frac{\partial t}{\partial x} = 0\,; \quad \frac{\partial \rho}{\partial x} = 0$$

This indicates that the shear stresses (equation (6)) are zero. By combining equations (2) and (8) it can then be written

$$\frac{\partial^2 \rho}{\partial r^2} + \frac{1}{r}\frac{\partial \rho}{\partial r} - \frac{\rho}{r^2} - \frac{m+1}{m-1}\alpha \frac{a}{r} = 0 \quad .. \quad (2a)$$

In this equation the time factor is eliminated and ρ and r are the only two variables. The partial differential equation (2) can now be written as an ordinary differential equation ($\partial = d$), and by combining the various constants of the equation in the constant p it follows

$$\frac{d^2 \rho}{dr^2} + \frac{1}{r}\frac{d\rho}{dr} - \frac{\rho}{r^2} - \frac{p}{r} = 0 \quad .. \quad (2b)$$

By integration we obtain

$$\rho = Ar + \frac{B}{r} + \frac{p}{2} r \,.\, \ln r \quad .. \quad (9)$$

where A and B are constants. By partial differentiation we obtain

$$\frac{\partial \rho}{\partial r} = A - \frac{B}{r^2} + \frac{p}{2}(\ln r + 1) \quad .. \quad .. \quad (10)$$

By substituting (2b), (9), and (10) in the equations (3), (4) and (5), we obtain

$$\sigma_x = \frac{2G}{m-2}\left[(m-1)k + 2A - \frac{(m+1)(m-2)}{(m-1)}\alpha a \ln r + \frac{m+1}{m-1}\frac{\alpha a}{2} + (m+1)\alpha b\right] .. \quad (11)$$

$$\sigma_r = \frac{2G}{m-2}\left[k + mA - (m-2)\frac{B}{r^2} - \frac{(m+1)(m-2)}{2(m-1)}\alpha a \ln r + \frac{m+1}{2}\alpha(a+2b)\right] .. \quad (12)$$

$$\sigma_p = \frac{2G}{m-2}\left[k + mA + (m-2)\frac{B}{r^2} - \frac{(m+1)(m-2)}{2(m-1)}\alpha a \ln r + \frac{m+1}{m-1}\frac{\alpha a}{2} + (m+1)b\alpha\right] .. \quad (13)$$

The constants A, B and k can be determined from the boundary conditions for σ_r, and from the expression for the total axial force

$$P_x = \int_{r_i}^{r_u} 2\pi r \sigma_x \, dr \quad .. \quad .. \quad .. \quad (14)$$

The equations (1) and (7) are therefore solved.

Application to the piston rod of a Diesel Engine.

A leading manufacturer of marine Diesels experienced the fracture of the piston rods of his engines. It was assumed that these fractures were due to high thermal stresses. The salient data of the double-acting two stroke cycle engine in question were :

Cylinder bore	600 mm.
Stroke	900 mm.
Engine speed	100 r.p.m.
External diameter of piston rod	200 mm.
Internal diameter of piston rod	80 mm.
Weight of piston	1000 kg.
Weight of piston rod ..	600 kg.
Ignition pressure	50 kg. per sq. cm.

From these data the mechanical stresses occurring in the piston rod section were found as

$$\sigma_x = -490 \text{ kg. per sq. cm.}$$

and $\sigma_x = +445$ kg. per sq. cm. respectively.

Temperature measurements carried out prior to the fracture of the piston rod had shown that the external surface had a temperature of 280 deg. C. on the average, while the water cooled internal surface had a temperature of 140 deg. C. Introducing these temperature values into equation (8) we find

$$140 = a \ln 4 - b \quad \text{and} \quad 280 = a \ln 10 - b$$

and therefore $a = 152{\cdot}79$; $b = 71{\cdot}82$

On the piston rod surfaces the radial stresses σ_r must be zero ; with $\alpha = 0{\cdot}000012$, $m = \frac{10}{3}$, $r_u = 10$, and $r_i = 4$ cm., equation (12) yields

$$k + \frac{10}{3}A = -0{\cdot}0020841$$

$$B = +0{\cdot}029714$$

The constants k and A can be found from equations (11) and (14), with $P_x = 0$, yielding $k = 0{\cdot}0027632$ and $A = -0{\cdot}0014542$.

Assuming $G = 810{,}000$ kg. per sq. cm., equations (11), (12), and (13) can now be solved, and we find

$$\sigma_x = 10{,}906 - 5{,}516 \ln r \quad .. \quad .. \quad (15)$$

$$\sigma_r = 6{,}832 - 2{,}758 \ln r - 48{,}137 / r^2 \quad .. \quad (16)$$

$$\sigma_p = 4{,}074 - 2{,}758 \ln r + 48{,}137 / r^2 \quad .. \quad (17)$$

This shows that $\sigma_x = \sigma_r + \sigma_p$

Also, considering that the mechanical stresses are

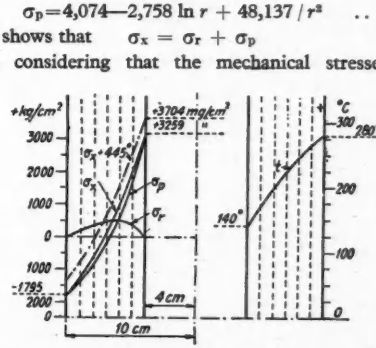


Fig. 2. Temperature and stress distribution in piston rod section of Diesel engine.

superimposed upon σ_x, it is found that the stresses are highest in the axial direction. Equation (15) yields the axial stress in the internal piston rod surface as

$$\sigma_x = 10{,}906 - 5{,}516 \ln 4 = +3{,}259 \text{ kg. per sq. cm.}$$

and in the external surface as

$$\sigma_x = 10{,}906 - 5{,}516 \ln 10 = -1{,}795 \text{ kg. per sq. cm.}$$

By adding the respective cyclic stresses of +445 kg. per sq. cm. and —490 kg. per sq. cm., we find that the stress in the inside surface of the piston rod varies cyclically between

$$\sigma_{max} = +3{,}259 + 445 = +3{,}704 \text{ kg. per sq. cm.}$$

and $\sigma_{min} = +3{,}259 - 490 = +2{,}769$ kg. per sq. cm.

These stresses are excessive for the material in question, and led to the fracture of the piston rod. Fig. 2 gives a graphic representation of the stress distribution in the piston rod.

NOTES ON THE INFLUENCE OF AUSTENITIC GRAIN SIZE UPON THE MECHANICAL, PHYSICAL, AND TECHNICAL PROPERTIES OF CARBON STEEL.

By G. DELBART and R. POTASZKIM. (From *Revue de Métallurgie*, Vol. 41, 1944. No. 4, pp. 97-110; No. 5, pp. 143-149).

EXTENSIVE research into the influence of the austenitic grain size upon the mechanical, physical, and technical properties of steel has been carried out in recent years. Although the problem has not yet been solved in its entirety, a good deal of progress has been made to be of value in numerous cases as they occur in practice. The investigations referred to below were made at the laboratories of the Société F.C.M., formerly Etablissements Cail, Denain, France.

Both the deoxidation methods and the compositions of the four heats investigated are given in Table I. All four heats were prepared in a Héroult electric arc furnace. Three of the heats were of low carbon content, while the fourth was a 0·4 C. material. The austenitic grain size was determined in accordance with the MacQuaid-Ehn test, the results of which are given in Table II.

Referring to Table II, it will be seen that ladle additions of 0·24 to 1·15 kg. of aluminium per ton resulted in a mixed grain size and normal structure. Increasing additions resulted in reduction in the size of both large and small grains and an increase in the relative amount of large grains present. Additions to the ingot mould as well as to the ladle resulted in a very fine grain of abnormal structure.

The test pieces of the low carbon material (Ingots B and D), which had been cast in rounds of 30 mm. diameter and then forged into bars of 15 by 15 mm. cross section, were partly annealed at 900 deg. C. for half an hour followed by cooling in air, the balance being subjected to the same kind of anneal and then to heating at 650 deg. C. for 2 hours with subsequent cooling in air. In the tensile test the fine-grained and the coarse-grained material in the annealed condition exhibited substantially identical tensile strength, the heat treated fine-grained material being slightly inferior in strength. But the elastic limit of the fine-grained material proved to be superior, irrespective of the heat treatment. In annealed condition elongation of the fine-grained material proved to be identical with that of the coarse-grained samples, while the heat treated fine-grained material gave a slightly higher elongation. With regard to the reduction of area, the fine-grained material was superior; but this superiority was more marked in the case of the heat treated samples.

Samples were carburized in boxes of $1{,}400 \times 600 \times 400$ mm., the carburizer being composed of equal parts of fresh and of recovered Houghton L compound. Carburizing time was 40 hours for normal penetration and 102 hours for deep penetration. Microscopic inspection revealed that the depth of penetration was virtually unaffected by the austenitic grain size. The depth of the hypereutectoid zone—after cooling the carburizing box and before heat treatment—was found to range from 0·7 to 0·8 mm. in the case of normal depth of penetration, that is to say, 47–57 per cent of the depth of case. In the pieces subjected to deep penetration, the hypereutectoid zone varied from 2·3 to 2·75 mm., that is, 61 to 72 per cent of the depth of case. After heat treatment the hypereutectoid zone was found to have disappeared.

It was found that carburizing time was without influence upon the number of soft spots. After quenching in tap water, soft spots were found to occur more or less in all steels excepting the 0·4 C. material. Generally, the prevalence of soft spots was greater in the carburized fine-grained samples. Salt water quenching was found to be a very efficacious means for suppressing the occurrence of soft spots even in the case of very fine grain material, which in the tap water quench developed a large number of soft spots. The average surface hardness, however, was found to be the same regardless of whether the quench was carried out in tap water or salt water.

Regarding hardness variation in the depth of the case, it was found that with one single exception, the maximum hardness of the fine-grained materials was less than that of the coarse-grained samples, the difference in maximum hardness averaging some 0·5 to 2·0 Rockwell units. Furthermore, the maximum case hardness of the fine-grained steel was found to occur closer to the surface; and it was also established that the difference in surface hardness of the fine-grained and the coarse grained materials is substantially in proportion with the difference in maximum hardness. Moreover, the case of the fine grained material is more shallow.

NOTCHED BAR IMPACT TESTS.

Notched bar impact tests were conducted on carburized samples of 160 mm. length and of 30×30 mm. section, the notch ending in a hole of 4 mm. diameter. The depth of the case was 1·7 mm., and 1·6 mm. in the steels *D* and *B* respectively. From the impact tests the following conclusions were drawn:

The fine grained steel has the greater impact strength. Duplex hardening has a favourable effect upon the bending strength of the material, particularly of the coarse grained steel; but it adversely affects the notched bar impact strength of both the coarse-grained and the fine-grained steels. Increased impact strength is characterized by a fracture which shows a fine texture in the core. Near the surface the fractures of the duplex hardened samples had a markedly silky texture, which does not, however, appear to be related to the improvement in impact strength. The latter appears to be related rather to ferrite precipitations, the impact strength increasing with a diminution in such precipitation.

In heating the samples to 900 deg. C. prior to the quench, the material was protected against decarburization by packing it in used carburizer, while reheating to 770 deg. C. was carried out in the furnace atmosphere. Inspection of the quenched material revealed the presence of spots of troostite in the carburized case. These spots may have been due to

TABLE I.—Casting Details

Ingots at 500 kg. each (275 × 230 mm).

No. of Heat	Ingot Mark	Charge			Desoxidation				Aluminium Additions		
						Heat		Ladle	Ladle	Mould	Total
		Sheet Iron kg.	Scrap kg.	Total Charge kg.	Hemat. per cent.	Fe Mn at 80 per cent. per cent.	Fe Mn at 45 per cent. per cent.	Fe Si at 90 per cent. per cent.	kg./ton	kg./ton	kg./ton
46,472	D	1,700	4,100	5,265	—	0·66	—	0·10	0·240	—	0·240
	B								0·240	0·350	0·590
46,475	C	1,700	4,100	5,250	0·38	0·61	—	0·10	0·530	—	0·530
46,478	A	1,700	4,000	5,135	0·58	0·74	—	0·10	1·150	—	1·150
46,481	E	1,500	3,700	4,665	4·3	—	1·6	0·23	0·270	—	0·270
	F								0·270	0·250	0·520

TABLE II.—Aluminium Additions *versus* Grain Size

No. of Heat	Ingot Mark	Aluminium Additions gr/t			Austenitic Grain Size			Structure
		Ladle	Mould	Total	A.S.T.M. Grain Size	Mean Surface in μ^2	Type of Grain	
46,472	D	240	—	240	33 % No. 1 + 67 % Nos. 4-5	8,000	1/3 very large + 2/3 medium	Normal
46,472	B	240	350	590	Nos. 7-8	600	Very Small	Abnormal
46,475	C	530	—	530	47 % No. 2 + 53 % No. 5	7,300	1/2 large + 1/2 medium	Normal
46,478	A	1150	—	1150	72% Nos. 2-3 + 28% Nos. 5-7	5,000	5/7 large + 2/7 medium	Normal
0·40 per cent Carbon Steel.								
46,481	E	270	—	270	40 % No. 2 + 60 % No. 5	6,000	2/3 large + 1/3 medium	Normal
46,481	F	270	250	520	No. 8	500	Very small	Abnormal

0·10 per cent Carbon Steel.

TABLE III.—Influence of Austenitic Grain Size upon Critical Points

No. of Heat	Method of Cooling	Ac_1			Ac_3			Ar_1			Ar_3			$Ac_1 - Ar_1$			$Ac_3 - Ar_3$		
		Large Grain	Small Grain	Difference in deg. C.	Large Grain	Small Grain	Difference in deg. C.	Large Grain	Small Grain	Difference in deg. C.	Large Grain	Small Grain	Difference in deg. C.	Large Grain	Small Grain	Difference in deg. C.	Large Grain	Small Grain	Difference in deg. C.
46,472	Air	724	727	+ 3	852	866	+14	550	555	+ 5	762	782	+20	174	172	− 2	90	84	− 6
	Furnace	724	727	+ 3	852	866	+14	632	647	+15	797	817	+20	92	80	−12	55	49	− 6
46,481	Air	725	729	+ 4	794	800	+ 6	520	540	+20	672	700	+28	205	189	−16	122	100	−22
	Furnace	725	729	+ 4	794	800	+ 6	612	627	+15	713	733	+20	113	102	−11	81	67	−14

particles of carburizer which had not been removed from the surface of the material prior to the quench. In contrast to this, the surface of the samples quenched from 770 deg. C. proved to have a uniform martensitic structure, which might account for the smaller impact strength of the material.

The influence of the austenitic grain size upon the transformation point was also studied, the test results being listed in Table III. It was concluded that

(a) The critical points Ac_1, Ac_3, and Ar_1, and Ar_3 are higher in the fine-grained material.

(b) The respective Ac_1 points shows the smallest divergence (3 to 4 deg. C.) and the respective Ar_3 points show the widest divergence (20 to 28 deg. C.).

(c) The transformation lags in cooling Ac_1—Ar, Ac_3—Ar_3 are always smaller in the fine-grained materials.

Forgeability tests were conducted on bars of 9·77 mm. diameter tested to destruction in the tensile test at 800, 900, 1000, and 1100 deg. C. The shape of the fractured pieces showed that grain size exerts a considerable influence upon hot ductility. While the coarse-grained samples were forgeable at a temperature as low as 900 deg. C., the fine-grained material requires at least a temperature of 1100 deg. C. Also, forgeability diminishes with the amount of aluminium used in the deoxidation process; or, in other words, forgeability increases with the grain size. Thus, the steel *D*, which had received an addition of 240 grams of aluminium per ton in the ladle exhibited the best forgeability, while the steel *A* with an addition of 1,150 grams of aluminium per ton in the ladle was much less easily forged. Medium forging qualities were

TABLE IV.—CREEP TESTS

Test bars: dia. = 4 mm, l = 100 mm and 32 mm made from bars of 20 mm dia.

Heat treated
Annealed at 850 deg. C., cooled in furnace

No. of Heat	Stress in kg/mm²	Elongation in 10^{-3} per cent.				Creep rate in 10^{-4} per cent per hour between		
		Initial	Total after 45 hours	Recovery	Permanent after 45 hours	3rd–6th hour	5th–10th hour	25th–35th hour
				400 deg. C.				
46,481 E coarse grain	15	110	144	88	56	11·0	8·4	3·4
	18	138	182	99	83	16·6	12·0	4·8
	20	—	—	—	—	31·2	23·1	9·3
	21	255	380	128	252	46·0	31·0	14·6
46,481 F fine grain	10	77	116	70	46	13·3	11·0	4·0
	12	92	166	74	92	25·8	21·8	7·8
	20	545	—	—	—	451	300	86·3
				450 deg. C.				
46,481 E coarse grain	11	50	106	66	40	15·0	11·0	4·2
	14	—	—	—	—	28·1	18·7	7·5
46,481 F fine grain	4	23	32	23	9	—	—	1·5
	6	78	122	53	69	17·7	12·5	7·8
	8	75	142	65	77	23·3	17·0	9·8
	14	—	—	—	—	—	—	164

No. of heat	400 deg. C.		450 deg. C.	
	Creep strength in kg./mm² for a maximum creep rate of:—		Creep strength in kg./mm² for a maximum creep rate of:—	
	5×10^{-4} per cent/hour between 25th and 35th hour	10×10^{-4} per cent/hour between 25th and 35th hour	5×10^{-4} per cent/hour between 25th and 35th hour	10×10^{-4} per cent/hour between 25th and 35th hour
46,481 E (coarse grain)	18·1	20·2	12·5	15·5
46,481 F (fine grain)	10·5	12·5	5·5	8·0

found to obtain with steel *C*, which had received an addition of 530 grams of aluminium per ton.

CREEP RESISTANCE.

Creep tests were conducted on bars of 4 mm. diameter and of 100 and 32 mm. length respectively, made from 20 mm. dia. bars. The test results obtained at 400 and 450 deg. C. respectively, based on test runs of 45 hours each, are listed in Table IV.

From these data it will be seen that the coarse grain material possesses a considerably greater creep strength, the difference in its favour as compared to the fine-grained material being some 70 to 80 per cent at 400 deg. C., and some 90 to 125 per cent at 450 deg. C.

THE INFLUENCE OF SHAPE UPON THE STRENGTH OF A BODY UNDER STRESS.

By A. THUM. (From *V.D.I.-Zeitschrift*, Vol. 88, November 11th, 1944, pp. 609-615).*

WHILE a first approach to the scientific analysis of the strength of a stressed body can be traced back to Galileo Galilei, it was Hooke, who in 1678, laid the foundation of the scientific theory of elasticity by his dictum 'ut tensio sic vis.' But even Hooke did not yet perceive the concept of tensile stress; this was established some 150 years later by the French mathematician Cauchy. It may appear surprising that such a long interval came to pass until the concept of stress was established, but this can be explained by the circumstance that the state of stress as such is of an abstract nature and cannot be measured directly, but can only be determined by computation. In the case of complex designs it cannot even be arrived at by computation, but must be determined from the measured strain.

Beyond all doubt, the theory of elasticity and the analytical methods based upon it were responsible for outstanding achievements and advances. In fact, until some twenty years ago, the elastic theory formed the basis of machine design as a whole. However, it would seem that inexplicable failures of design gradually led to the conclusion that the theory as such possessed far-reaching defects. Frequent breakdowns of machinery and fractures of component parts occurred when the tendency set in to increase the physical size of machines beyond what had been previously attempted. Striking examples of the difficulties encountered in this respect can be found in the development of steam turbines, boilers, of the Diesel engine, electrical machinery, and also in the field of high speed traction and aviation.

Unaware of the true cause of a failure the designer would attempt to purchase renewed safety by simply increasing the physical dimensions of the piece that had suffered fracture. But often this did not remedy matters, and in fact it was generally the case that the increased size only aggravated the danger of failure. These difficulties led in many cases to the abandonment of inherently sound developments; they also gravely impaired confidence in the abilities of the designers concerned. But more often than not such disappoint-

* Abstract of a lecture given to a meeting of the V.D.I. on 28th August, 1944, on the occasion of conferring to the author the Grashof medal, the highest German distinction for engineers.

ments were due not to a lack of ability on the part of the designer, but to a one-sided overestimation of the applicabilities of existing methods of analysis and design. In brief, no cognizance had been taken of the fact that a knowledge of elastic theory, statics, dynamics and thermodynamics does not suffice to solve all the problems encountered in design technique.

Indeed, the real cause of these shortcomings can be said to be lack of attention paid to the technological aspects involved in machine construction and lack of knowledge with respect to the behaviour of materials under operating conditions. All computations of the physical phenomena involved in a given machine will not avert failure if unsuitable materials are chosen and if the stressed part is not given an appropriate shape. Consideration must also be paid to the fact that the materials employed in the construction of machines are neither homogeneous nor ideal, but that they possess imperfections due to the inevitable factor of human error in their production. Moreover, each piece of material, as for instance, a steel shaft or a piece of steel wire, has a history insofar as it retains the characteristics impressed upon it by the various manufacturing processes through which it has passed. This history must be taken into account by the designer in order to evolve a successful design.

Some twenty years ago a growing recognition of these problems led to the establishment of laboratories devoted to technological research. Also, technological knowledge had become greatly enriched by the findings of research workers in the newly created sciences of metals and metallography. But even so, design technique still remains afflicted with the fundamental shortcoming that computations are based upon the elementary concepts of stress and strain which are applicable merely to the simplest of structural shapes such as plain cylindrical or prismatic rods, spheres, cones, plates, discs, rings, etc. Computation of the stress distribution in the cross-sectional area of the piece is based on the assumption that tensile and compressive stresses are uniformly distributed and that torsion and bending stresses increase according to a linear characteristic. However, the unit stresses arrived at in this manner can only be considered as fictive or nominal stresses, as the uniform stress distribution assumed is hardly ever realised in practice. Also, notch-effect may be present, leading to considerable local stress concentrations far in excess of any nominal stress determined by computation. Notch-effects also occur at points of abrupt changes in cross-section, and at all such points where changes in the direction of stress occur.

Elementary elastic theory does not yet take account of the stress-raising influences of notches, as attempts at their analytical treatment lead to highly complex partial differential equations which cannot yet be solved. But it is these local stress increases which are especially harmful to fatigue strength. In the absence of analytical knowledge as well as of practical experience, a marginal coefficient—euphemistically called " safety coefficient " —was introduced ; but in many cases this turned out to be a coefficient of lack-of-safety rather than of safety, the greatest danger of fracture occurring where according to computation the factor of safety was held to be greatest.

It gradually came to be recognized that the neglect of the influence of shape upon strength was the main obstacle to progress. For this reason present-day design technique is no longer based upon strength values obtained from the testing of simple plain test bars, but careful consideration is paid to the influence which the shape of the stressed piece exercises upon the strength of the piece as a whole. The task of this new technique is, therefore, to evolve shapes in which stress concentration is eliminated to the greatest possible extent, and at the same time to reduce the dimensions at points of low stress to the minimum required. In other words, this technique aims at the design of uniformly stressed parts as far as this can be reconciled with other considerations of design.

This new method has proven particularly successful in cases where fluctuating or alternating stresses are to be carried. While in the old method it was considered sufficient to ascertain the so-called nominal stress, the new approach requires that the true stresses should be determined ; and their determination should not be limited to finding the stress prevailing at the weakest point, but should cover all points under stress. This demand is based upon the observation that with alternating or fluctuating load the highest stress does not always occur at the weakest section of the piece under stress, but that regions of stress concentration are places of incipient fracture. This phenomenon is strikingly illustrated by the fracture of identical pieces at widely separated points according to whether fracture is caused by static or dynamic loading. Once the point of greatest stress has been ascertained in a given design, the dimensions of other sections of the component under consideration can be reduced to a point where they are subjected to a stress as large as that prevailing at the critical point.

Attempts to determine actual stress distribution by mathematical means are more often than not futile. In fact, the analytical approach cannot even be recommended, since in the mathematical method certain interrelationships are lost sight of only too easily. The designer—and this is a most important maxim of the new method—must, therefore, attempt to anticipate what the probable stress distribution will be, making the most of his intuition.

Modern methods for measuring actual stress distribution are based on the measurement of local strain. Precision instruments have been developed which permit the measurement of local strain as small as 1/500 micron, so that stresses can be determined with an accuracy of 0·1 kg. per sq. mm. Owing to the small size of these instruments, they can be easily applied to narrow notches, and they can also be introduced into bores exceeding 12 mm. in diameter.

Stress concentration is expressed by the " shape factor " a_k, which represents the ratio of the maximum stress to the nominal stress. As inferred by its very name, the shape factor depends solely upon the shape of the stressed piece and not upon the material as such. For a number of simple shapes, numerical values of the shape factor have been determined by test and are available in tabular form. Where local stress concentration cannot be avoided by choosing a more appropriate shape, counteracting internal compressive stresses can be set up by such methods as local carburising, nitriding, or flame or induction hardening of the surface. The effects of impact stresses can be mitigated by imparting to the affected part a certain amount of elasticity by reducing its cross-section. With regard to the employment of high strength steels, it should be noted that their fatigue strength is affected much more by notch effects than is the case with ordinary steels. In order to be able to take full advantage of the greater fatigue strength of high strength steels, special care must, therefore, be taken to eliminate any possible notch effects.

In cases where low weight is of preponderant import, the component parts which carry the highest stresses occurring in the assembly concerned (automotive engines, airplane construction, etc.) are now frequently dimensioned on the time-strength principle, that is to say, their dimensions are reduced to a point where the component part is expected to fail after a given number of stress-cycles. This time-strength can be predicted from the N/S curve of the fatigue test. It goes without saying that component parts designed on this basis must be replaced after a requisite length

of operational service, in order to prevent their fracturing in service. This method appears as particularly attractive in cases where overloads are so few and so far between that it would not be justifiable to increase the dimensions, and with them the weight of the part, to a point where it possesses sufficient strength to withstand such occasional overloads for an indefinite length of time.

ROLLING OF SCREW THREADS.

By H. DINNER and W. FELIX. (From *Technische Rundschau Sulzer*, No. 1, 1945, pp. 131-134).

IN the works of Sulzer Bros., Winterthur, Switzerland, precision screw threads are frequently made by the thread rolling process. The superiority of a rolled screw thread over a machine-cut thread is due to its high accuracy, uniformity, and mirror-finish. Also, the manufacturing time of rolled threads is considerably shorter.

A number of tests were made to establish the characteristics of rolled threads under static as well as dynamic stress. The material employed in these tests was an open hearth bolting material of the following physical characteristics.

Modulus of elasticity ..	20,200 kg. per sq. mm.
Limit of proportionality (permanent set of 0·005 per cent)	29·0 kg. per sq. mm.
Yield point (0·2 per cent) ..	29·8 kg. per sq. mm.
U.T.S.	47·8 kg. per sq. mm.
Elongation (1=5d)	26·7 per cent
Reduction of area	69·0 per cent
Notched bar impact strength	20 mkg. per sq. cm.

In order to obtain comparative data, specimens of 1 in. thread were produced by (a) Rolling (b) Cutting (lathe) (c) Grinding (d) Milling.

The test bar dimensions are shown in Fig. 1. The static tensile and bending tests yielded the following results.

Thread produced by	Tensile test: Y.P. kg./sq. mm.	Tensile test: U.T.S. kg./sq. mm.	Tensile test: Type of fracture	Bending test: Angle of bend deg.	Bending test: Inner radius of bend mm.	Remarks
Rolling	38·2	57·8	homogeneous	160	8	fractured
Cutting	35·7	54·2	,,	180	1	incipient fissure
Grinding	34·7	54·4	,,	180	1	,,
Milling	33·4	53·0	,,	180	1	,,

In all cases the fracture occurred in the threaded part of the piece. Dynamic tensile tests were carried out by the pulsating stress method, subjecting the piece to 3 million load cycles at the rate of 350 cycles per minute. The results were as follows :

Threading carried out by	Fatigue strength, kg./sq. mm. (pulsating load)		
	Test bar Fig. 2a	Test bar Fig. 2b	
Rolling	29.2	20.0	20.0*
Cutting	17.2	12.0	—
Grinding ..	18.5	—	—
Milling	19.2	12.0	9.0*

*Specimen aged at 250 deg. C.

All pieces fractured on the threaded part in the nut. Vickers micro-hardness tests registered much greater variations in hardness throughout the rolled thread than in the threads produced by the other methods. As will be seen from Fig. 3 the hardness in the rolled thread varied from 148 Vickers to 218 Vickers. This, of course, was due to the considerable compression of the material caused by the thread rolling process.

Microscopic inspection was carried out on longitudinal sections. Referring to Figs. 4a-b the machined threads are seen to have normal microstructure ; while the rolled threads shown in Figs. 5a-b show a greatly deformed structure of the grain. Fig. 5a also shows that a cleavage exists in the top of the thread which is due to the upward displacement of the material when forming the thread by rolling. This cleavage is particularly pronounced at the first and the last threads where the material is able to flow in axial direction ; and this phenomenon is responsible for the fact that the first and the last threads are not fully formed. The circumstance that threading by rolling leads to a lengthening of the work piece is of no significance in most cases.

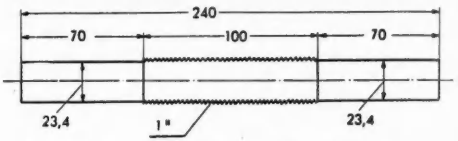


Fig. 1. **Dimensions of test bar for static tensile test.**

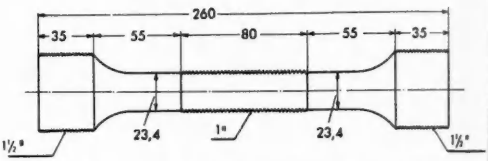


Fig. 2a. **Test bolt for pulsating stress test.**

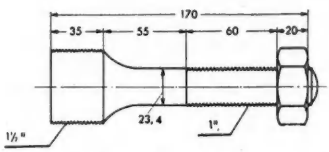


Fig. 2b. **Test bolt for pulsating stress test.**

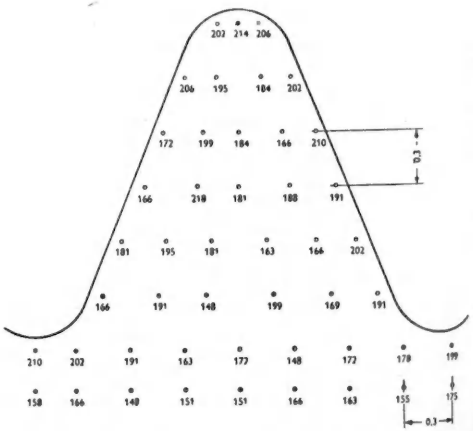


Fig. 3. **Vickers micro-hardness of rolled thread.**

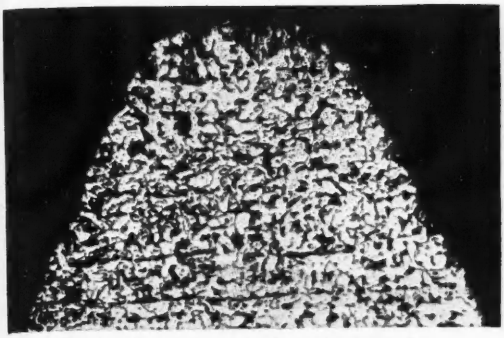


Fig. 4a. **Grain structure of machined thread.** (Top) Magnification 50×

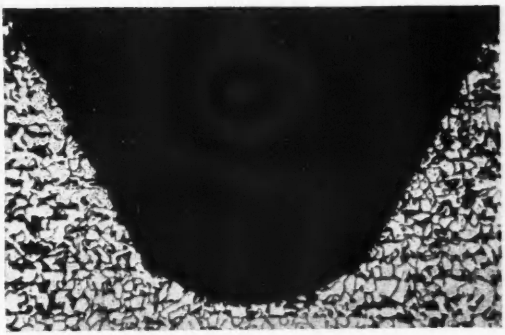


Fig. 4b. **Grain structure of machined thread.** (Root) Magnification 50×

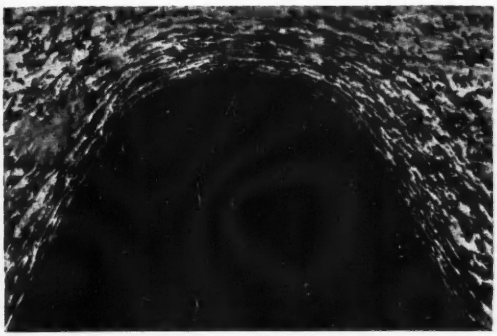


Fig. 5a. **Grain structure of rolled thread.** (Top) Magnification 50×

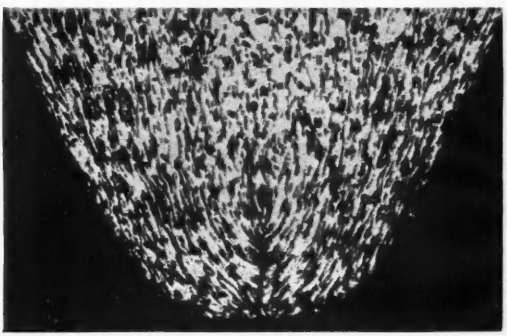


Fig. 5b. **Grain structure of rolled thread.** (Root) Magnification 50×

Summing up it may be said that the static tensile and bending strengths of rolled and of machined threads are practically identical. But in the case of dynamic loading a rolled thread offers the advantage of a greater fatigue strength. Also a rolled thread is not subject to ageing.

NEW METHOD OF IMPEDANCE MATCHING IN RADIO-FREQUENCY CIRCUITS.

By G. GUANELLA. (From *The Brown Boveri Review*, September, 1944, pp. 327-329).

THE impedances of the individual circuits of radio-frequency equipment are frequently unequal. In order to obviate the reflections and losses involved by mismatching, special matching devices have to be inserted between such dissimilar circuits for the transmission of energy. For instance, matching is necessary between the tubes of a transmitter output stage with high load resistance and the low-impedance antenna transmission line or feeder system. In the case of low frequencies transformers with a corresponding turns ratio can be employed. By reason of the unavoidable leakage inductance of the coupled transformer coils, high frequencies generally involve tuning by means of additional condensers, and should the working frequency be varied, corresponding re-tuning is therefore entailed.

A new coupler which obviates re-tuning is shown in Fig. 1a. It comprises two superposed windings W_1 and W_2 separated by an insulating tube R. Given symmetrical currents i_1 (full-lined arrows) the magnetic fields produced by two closely-spaced superposed sections of conductor practically neutralize each other, i.e., the mutual inductance of two successive turns of a coil can be neglected, while it is possible to replace the two windings by two straight conductors having the same cross-section, length, and spacing as the two developed windings. This Lecher wire system is represented in the equivalent diagram (Fig. 1b) by the equivalent line A.

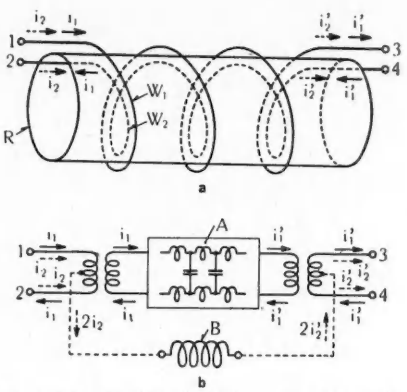
On the other hand, with unsymmetrical currents i_2 (dotted arrows), the field vectors produced by two superposed sections of the conductors are added together, with the result that the mutual inductance between the individual turns of the coil becomes an important factor. The double-wire coil system behaves here like a conventional choke coil, represented in the equivalent diagram by B. In this diagram the symmetrical and unsymmetrical currents i_1 and i_2, respectively, are segregated by centre-tapped ideal transformers T. Given an adequate number of turns on the windings W_1 and W_2 the impedance of the equivalent choke coil B becomes so high that, even assuming unequal potentials between the centre tappings of the input and output coils, the unsymmetrical current i_2 can be neglected. In this case the described coil system forms an ideal transformer combined with an ideal line.

In view of the effect of this ideal transformer such a system S can now be employed, as shown for example

in Fig. 2a, to couple a physically symmetrical circuit (connected to terminals 1 and 2) to a load resistance R_a having one pole earthed. By making the coil of suitable dimensions the surge impedance Z_o of the matching line (A in the equivalent diagram Fig. 1b) represented by the coil system can be adapted to the pure load resistance R_a. In this case the input impedance R_e occurring between terminals 1 and 2 is equal to the surge impedance Z_o and in consequence also to the load resistance R_a, immaterial of the actual working frequency.

By series-parallel connection of two or more coil systems impedance matching is also possible in a simple manner, independent of the frequency. Fig. 2b shows the input terminals of two systems of coils S connected in series and the output terminals in parallel. No objections can be raised to this practice provided the series inductance (B in the equivalent diagram in Fig. 1b) is large enough. The load resistance $R_a = \frac{1}{2}Z_o$ is thus transformed to the input impedance $2Z_o$. Analogously, with n coil systems impedance transformation in the ratio $1{:}n^2$ can be achieved.

In Fig. 2c, four coil systems are shown connected between a transmitter output stage and the high-frequency antenna cable K, the resulting impedance transformation being in the ratio $4^2{:}1 = 16{:}1$. With a coil system having a surge impedance $Z_o = 240\Omega$, for example, a transmitter output stage with a load impedance of $4 \times Z_o = 960\Omega$ can be coupled to an antenna cable of $Z_o : 4 = 60\Omega$. The coupled coil systems have the same effect as a transformer with separate windings, i.e., the symmetry of the anode circuit at the input end is not affected by single-pole earthing of the cable connected to the other end. Furthermore, the coupled coil systems behave like a Lecher wire system, i.e., the input impedance must follow a tangential function of the frequency when the terminals at the other end are open or short-circuited.



Figs. 1a and 1b. **Double-wire coil system with equivalent diagram.**

(a) The coil system comprises two superposed windings W_1 and W_2 separated by an insulating tube R (b). According to this equivalent diagram, where symmetrical currents i_1 are concerned, the coil has the effect of a Lecher wire system A, but with unsymmetrical currents i_2 the nature of a choke coil B. The symmetrical and unsymmetrical currents are segregated by ideal centre-tapped transformers.

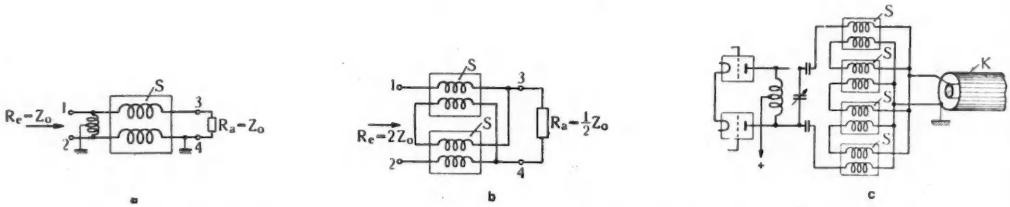


Fig. 2a, b, and c. **Employment of double-wire coil systems for coupling and impedance matching purposes.**

(a) Due to the suppression of the unsymmetrical currents by the series inductance of the coils such units can be used for coupling physically symmetrical circuits (connected to terminals 1 and 2) to circuits having one pole earthed (connected to terminals 3 and 4).

(b) By series parallel connection of two coil systems S the load resistance $R_a = \frac{1}{4}Z_o$ is transformed to the input impedance $R_e = 2\,Z_o$ (Z_o = surge impedance of a coil system).

(c) The antenna cable K and output stage are "matched" by the four coil systems S.
Surge impedance of coil systems $= 240\Omega$.
Surge impedance of cable
$= 240\ \Omega : 4 = 60\ \Omega$
Load impedance of output stage
$= 240\Omega \times 4 = 960\Omega$.

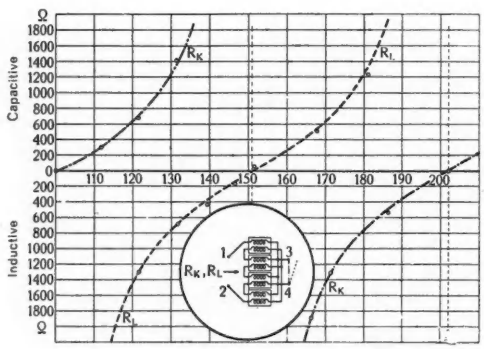


Fig. 3. **Input impedance of matching unit when output terminals short-circuited or open.**

The matching unit comprises four double wire coils in series-parallel connection. The computed and measured primary impedances are plotted as a function of the frequency with the secondary terminals open and short-circuited.

—·—·— } Computed curves o. Test points.

R_K. Input impedance with secondary terminals short-circuited.
R_L. Input impedance with secondary terminals open.

Fig. 4 gives the curve of the input impedance for a load impedance of about 53Ω. From the test points it is clear that the desired impedance transformation in the ratio of 1 : 16 is actually possible over a very wide frequency range. The deviation of the plotted mean-value curve R_e from the theoretical curve 1 is due to the load impedance being slightly lower than the theoretical value, as well as to the inherent capacitance of the circuit.

The described method of matching is particularly suitable for application in the ultra-short-wave field, where it represents a big simplification compared to conventional tuned matching devices.

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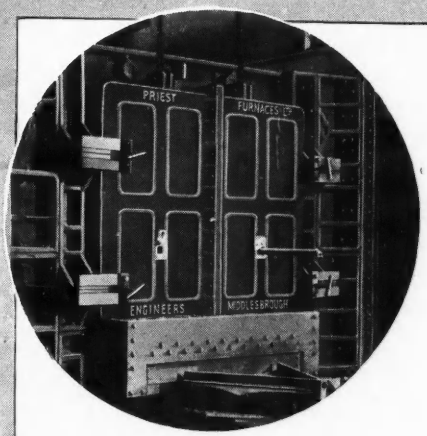

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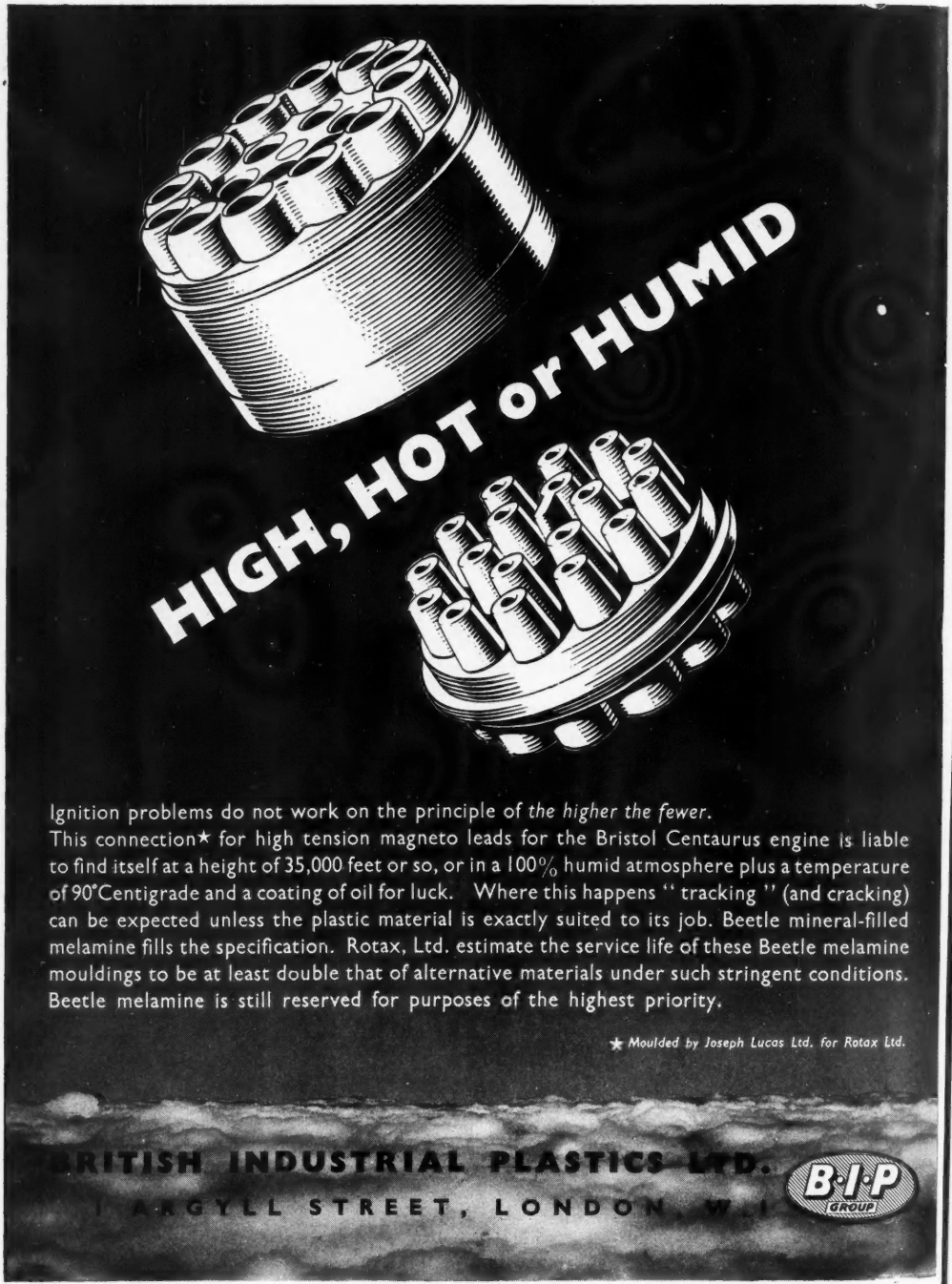

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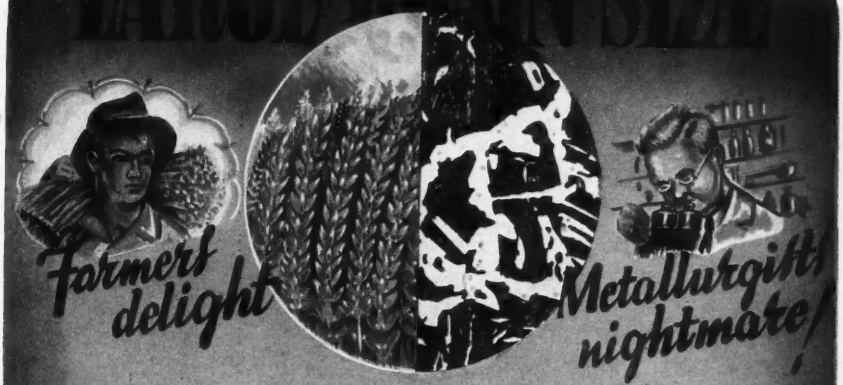

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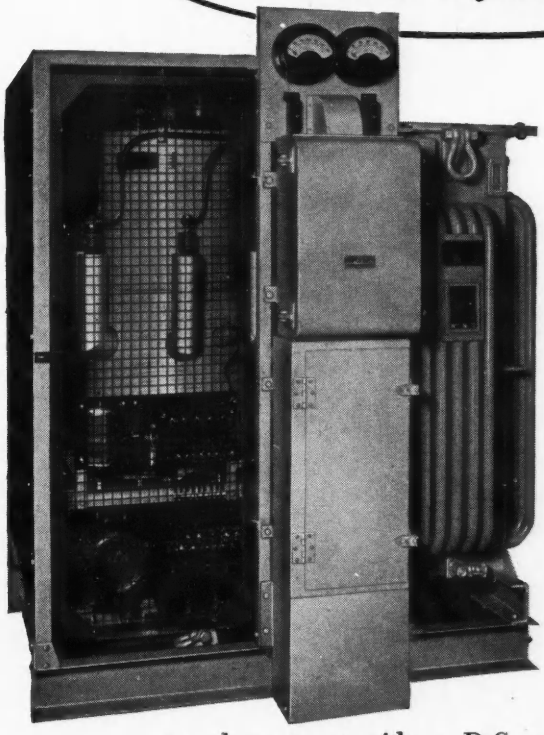

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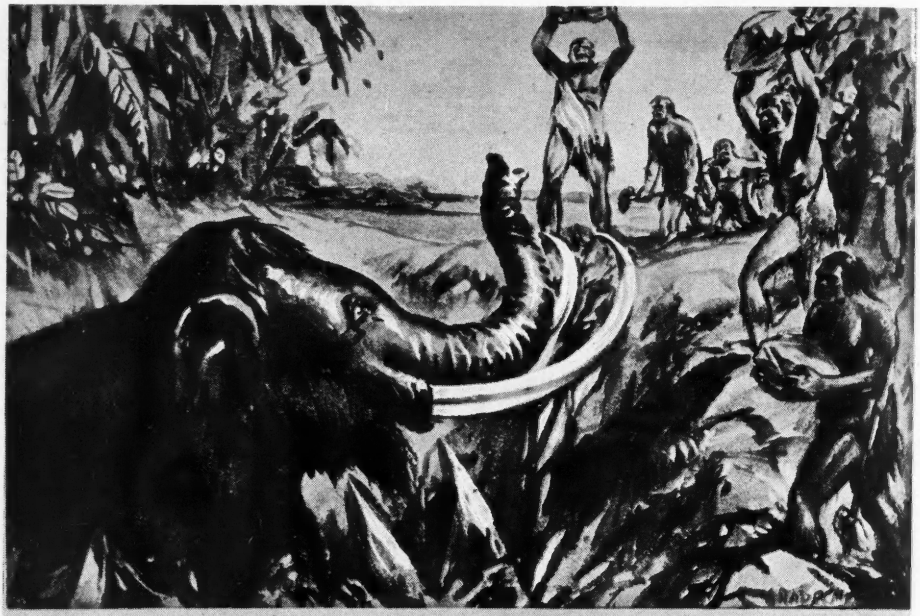

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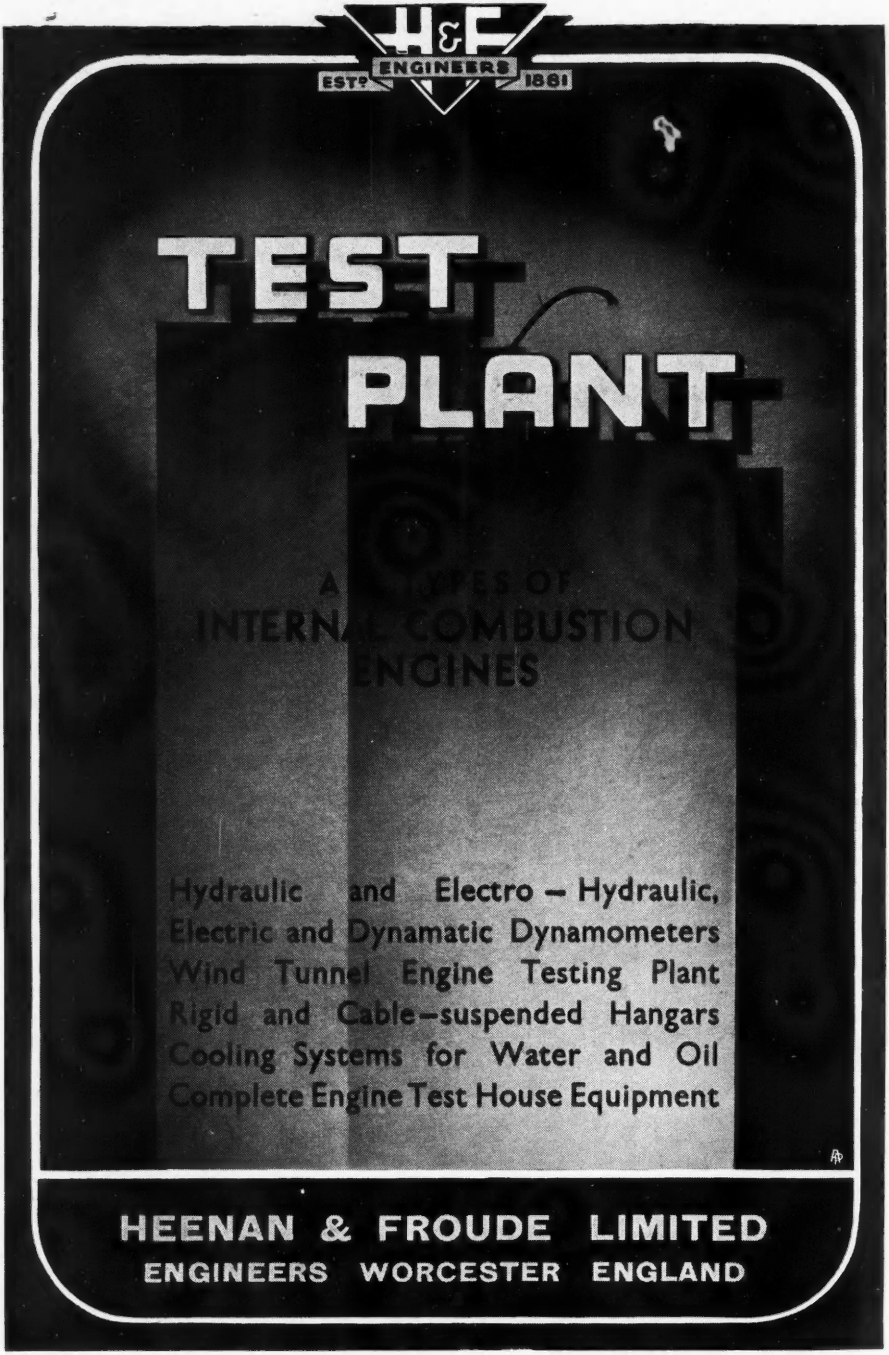
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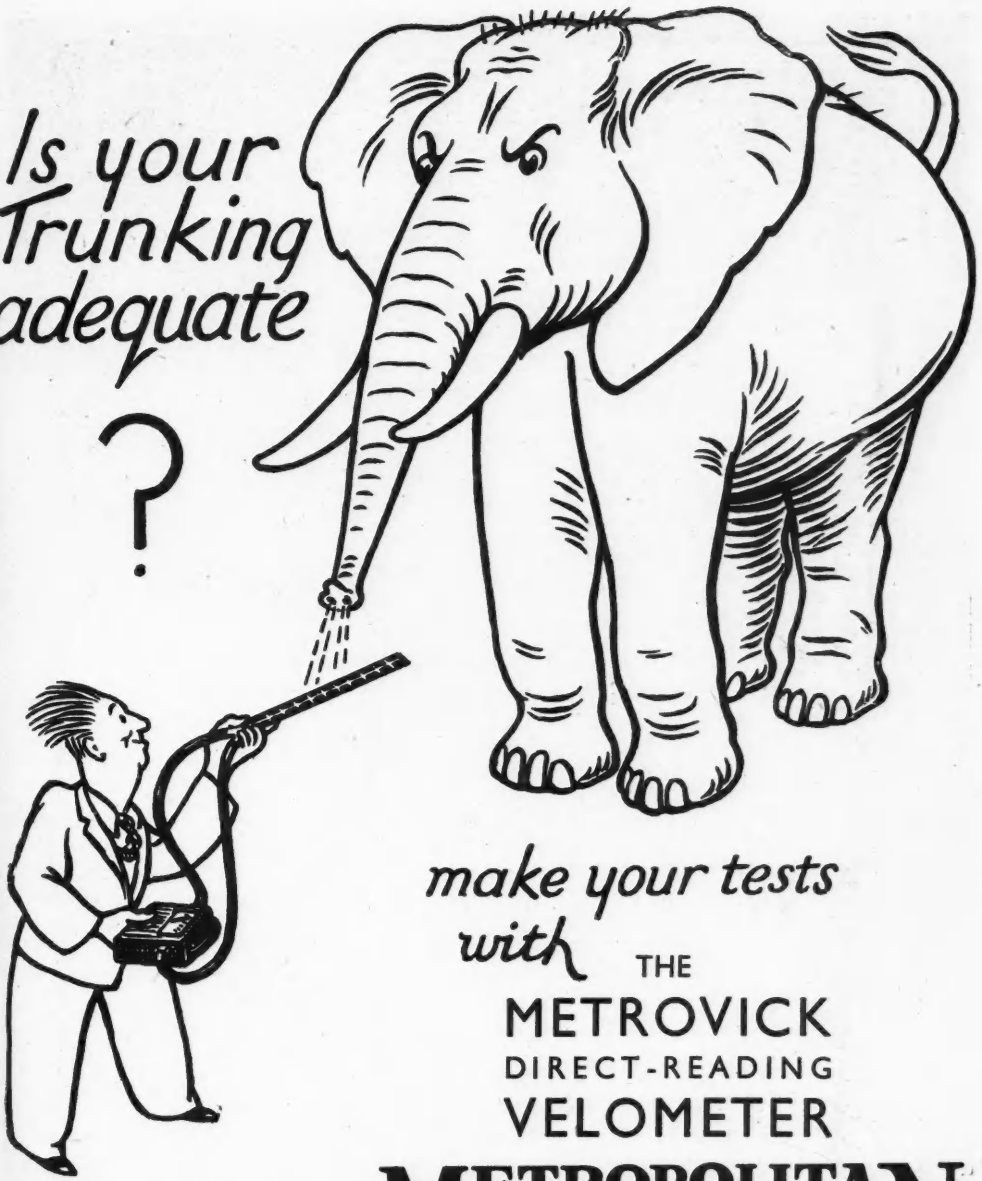

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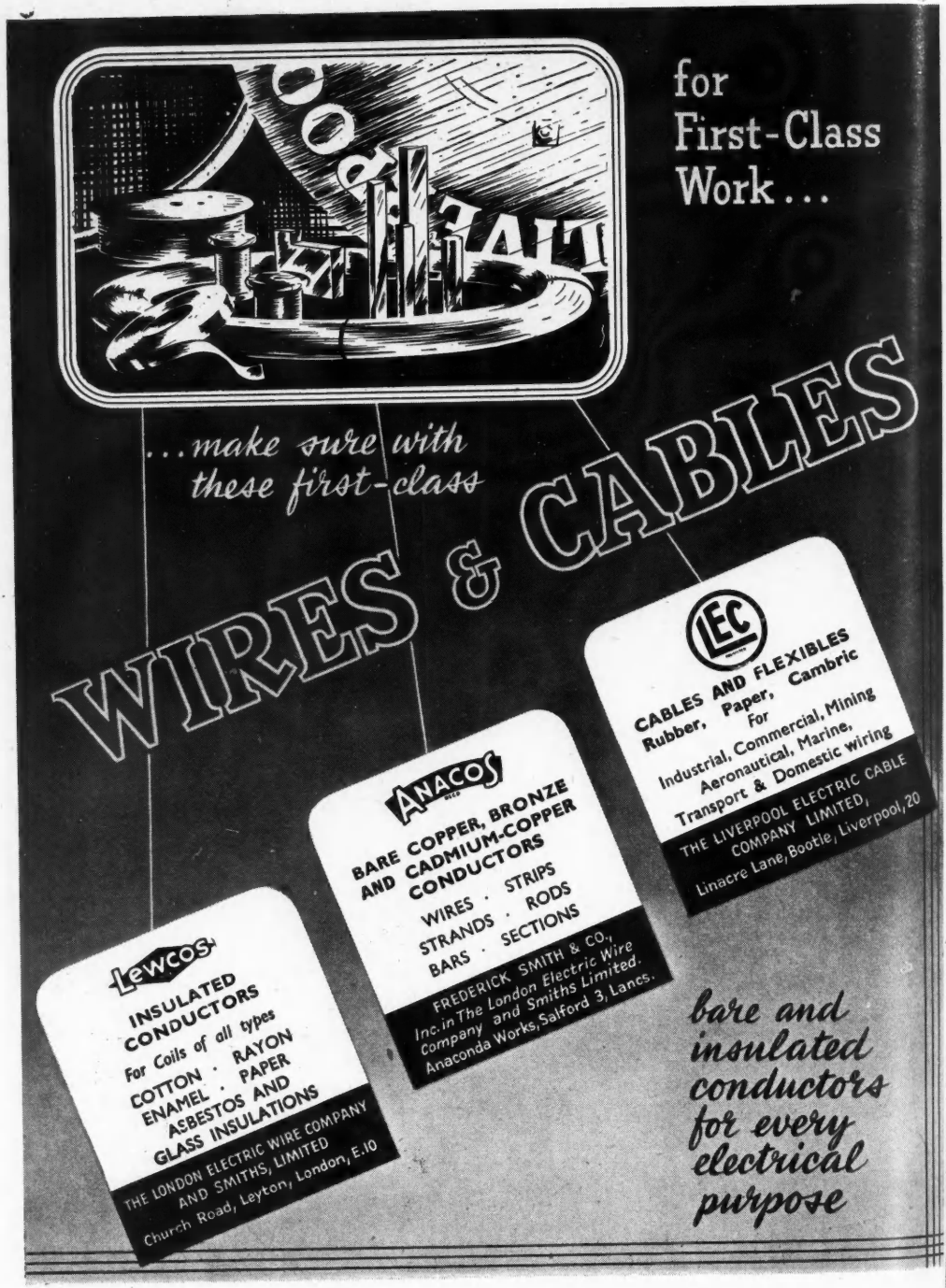
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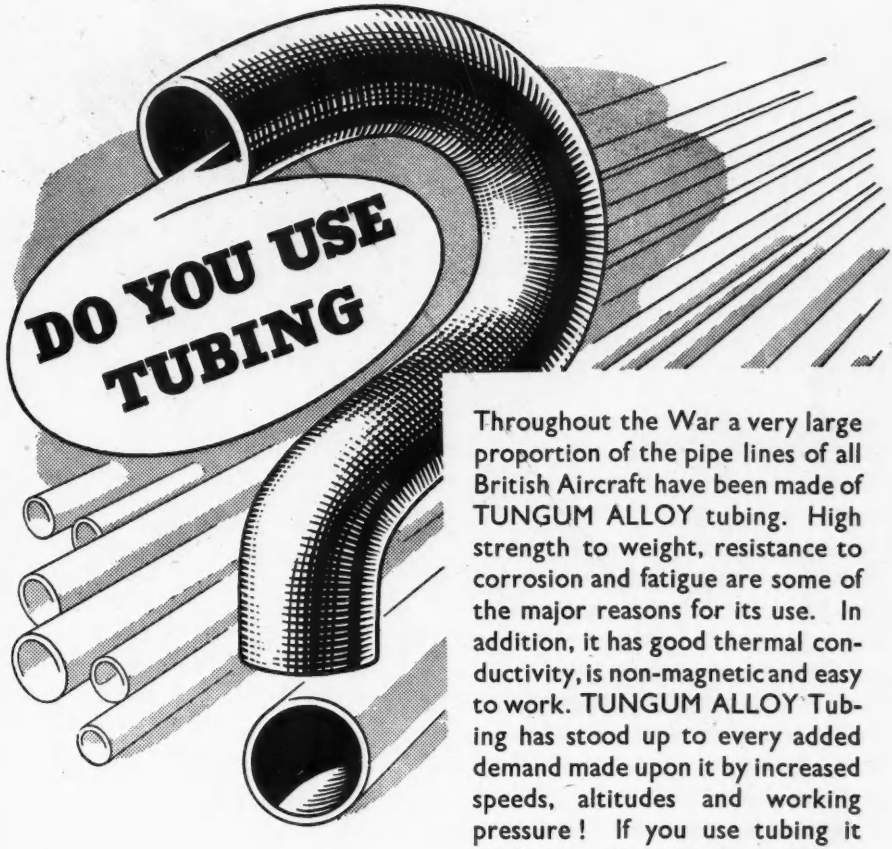

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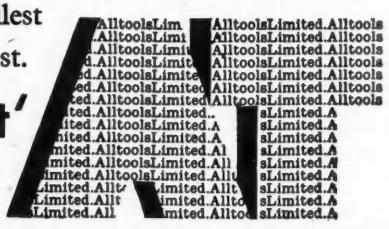

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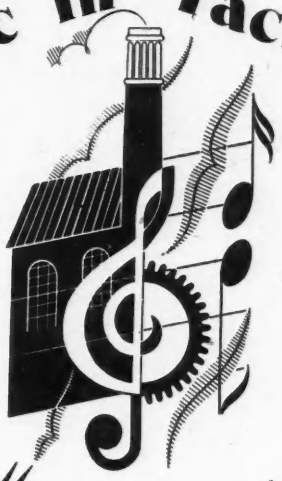

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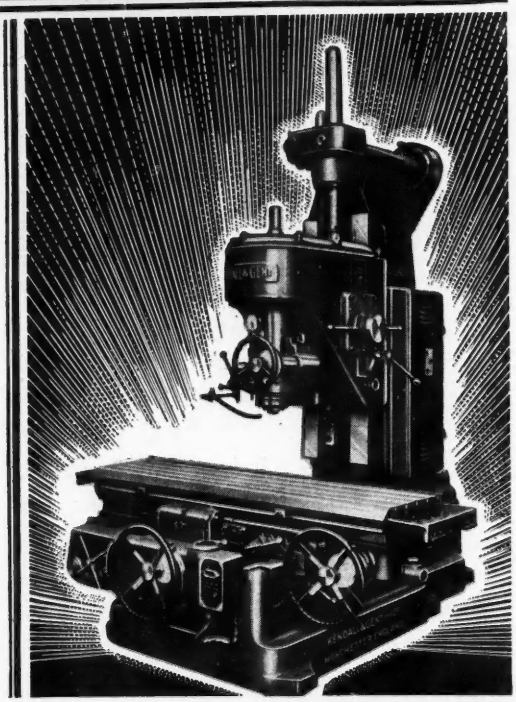

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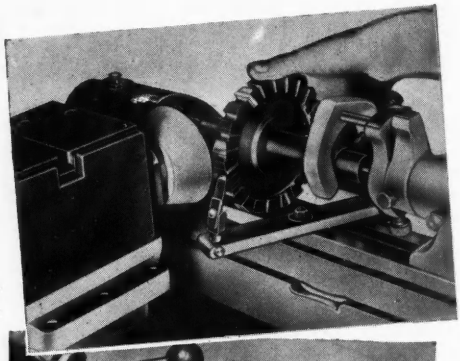

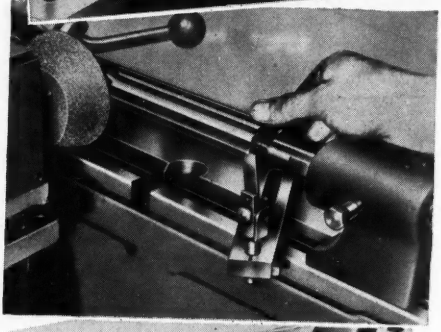

FIG. 301 8" × 16"

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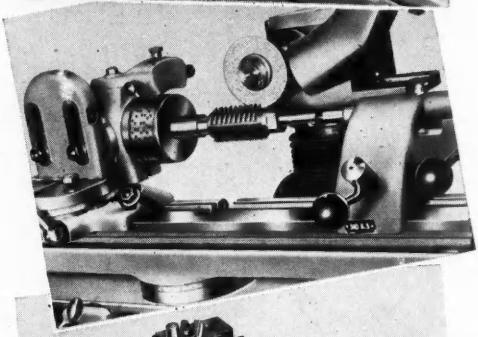
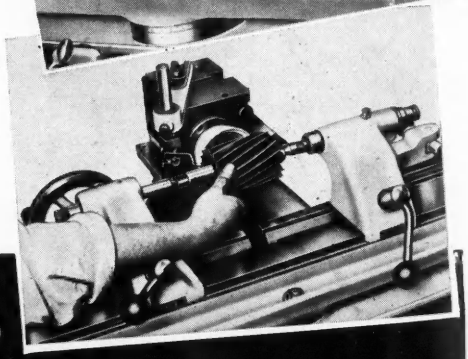
The Jones-Shipman Fig. 301 Cutter and Tool Grinder is designed to cover a wide range of operations in the shortest possible times, so that tools and cutters can be quickly and accurately ground.





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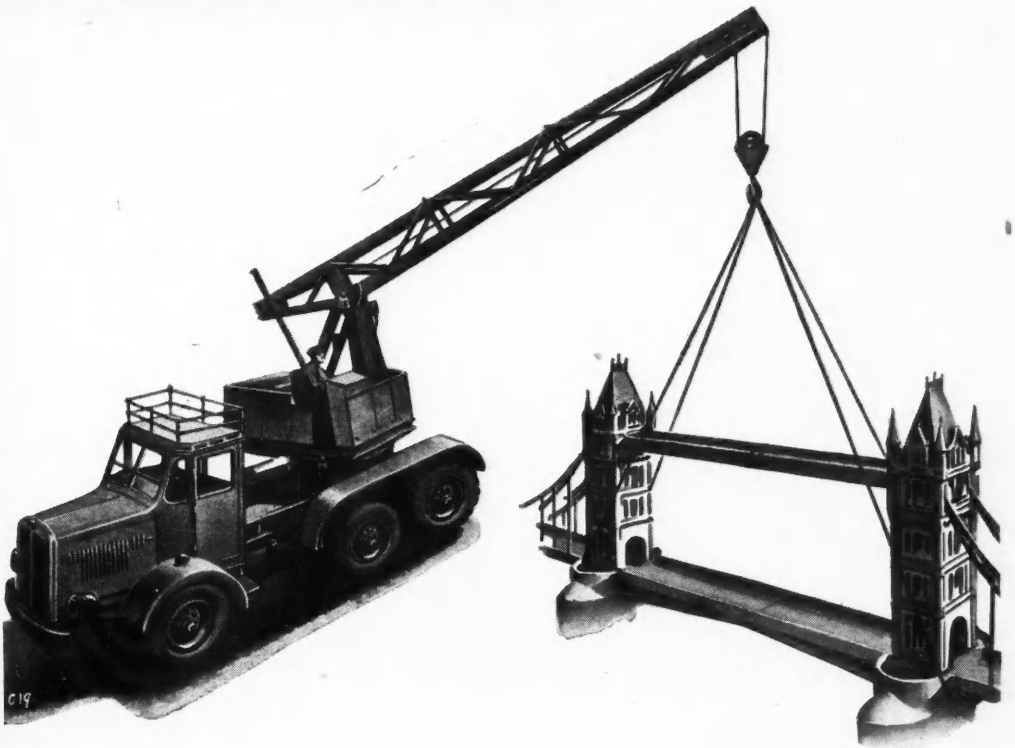

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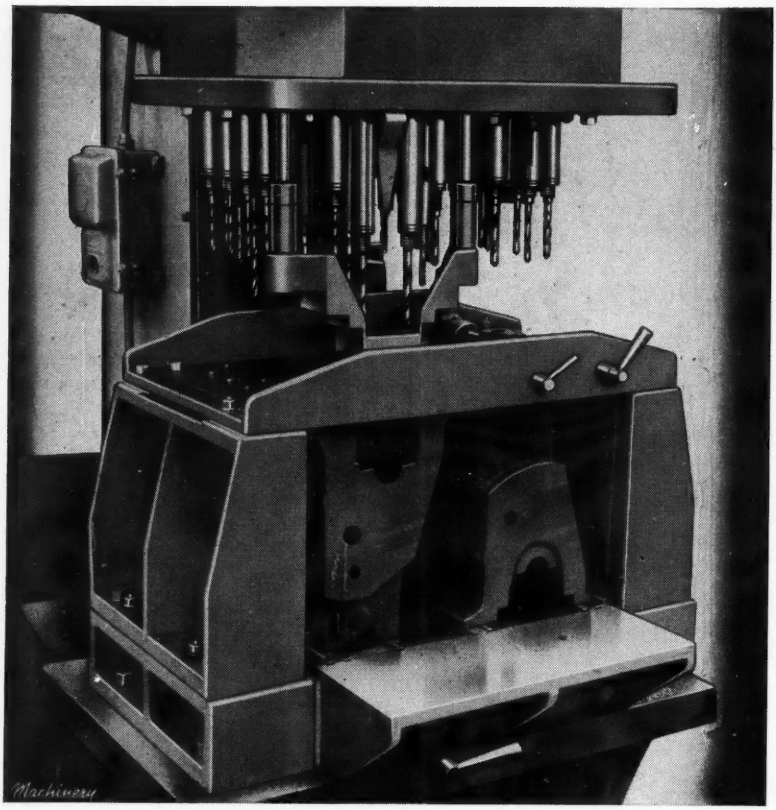

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Philidas engineers set to work to produce a stiffnut which would carry out its purpose, no matter what the odds. In order that no part of the bolt or the nut should wear unevenly, they decided that the stiffnut must depend on a homogeneous construction which would be equally effective in all types of metal. In order that it should be of universal application, the friction action in the threads must be capable of manufacture in any size and any thread, and the action of the friction portion cover several circumferences of the nut. Variations of temperature at extreme ranges must have no appreciable effect on the torque exerted on the bolt.

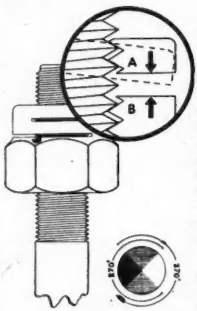
The Philidas stiffnut was specifically produced to meet these exacting conditions. It combined all the essential requirements.

After exhaustive tests, the Philidas was not only passed for unrestricted Service use but also recommended for tropical use. A fundamental advance in engineering design had been made. This new tested stiffnut overcame all the weaknesses which had previously been apparent.

All-round "pincer" grip

Wherever there is a problem that involves a constant grip on a threaded bolt, whether screwed to a face or unsupported, there is no doubt that it can be solved by the use of Philidas.

A "pincer" action is achieved on the bolt threads by making two parallel cuts from either side of the friction portion of the nut. The nut is then set. This results in threads of the friction portion being slightly out of phase. The torque produced on application is evenly spread over the complete circumference of several threads.



THE "PINCER" GRIP

The Philidas has two parallel interlocking cuts across the friction portion of the nut. After being set, the threads are slightly out of phase. When the stiffnut is applied, a considerable torque is produced, part of which is illustrated here. The flange A, attempting to return to normal, exerts a strong pressure down on the bolt thread, while the other arm of the "pincer" is made by the stable portion B.

This grip covers 270 degrees of the circumference of the bolt. The second cut, being diametrically opposite, exerts a similar force on the thread. The result is an even pressure, throughout the full 360 degrees, bearing on more than double the surface covered by any other similar type of stiffnut.

Despite the considerable torque exerted on the bolt, no damage is done to the bolt even after many hundred applications. Part of the torque tests included heating a sample mild steel stiffnut on a mild steel bolt to a temperature of 500 degrees C. The torque was measured at intervals of 50 degrees C. It was found that the torque never dropped appreciably below its initial value and was actually doubled at around 450 degrees C. At 500 degrees C., the limit of the test apparatus, the torque still remained well above its initial value.

Another outstanding advantage of the Philidas is that by reason of its homogeneous construction and its design, it can be made in any material, including the latest light alloys. For electrical purposes particularly, Philidas in brass has innumerable uses. If you have a vibration problem in your post-war production designs, Philidas will fix it for you.

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Six Philidas ¼-inch B.S.F. mild steel stiffnuts were submitted to exhaustive torque tests by the Air Ministry Testing Dept. Each nut was screwed on and off a mild steel bolt 50 times; the friction portion being completely disengaged each time. To pass the test the initial torque must never be less than 1.8 lb. ins.

In all cases the final torque was more than 50% of the initial torque.

The test was then continued in an attempt to destroy one of the nuts. After 1,000 applications the test was discontinued, as a torque of 1.3 lb. ins was still held (minimum test requirement 0.9 lb. ins) and showed no signs of reduction.

A projection of the bolt thread engaged by the friction head was examined for signs of wear during the test. Even after 1,000 applications, amount of wear on bolt thread was completely negligible.

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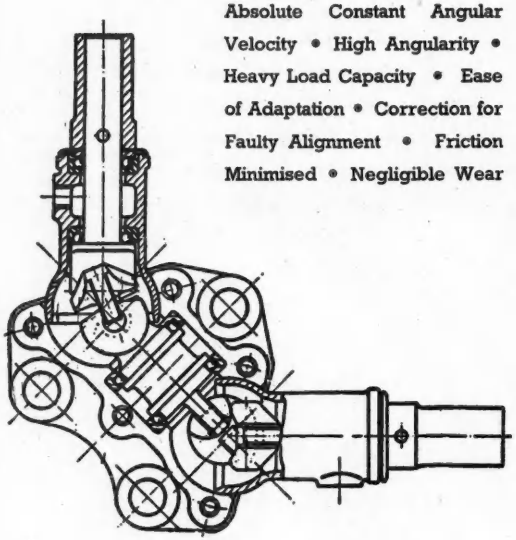

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(Note - All figures in parts per 100,000.)

Plant	No. 1		No. 2		No. 3	
Water	Crude	Treated	Crude	Treated	Crude	Treated
Cations						
Calcium Ca	3.2	–	9.4	–	10.7	–
Magnesium Mg	0.8	–	0.36	–	1.09	–
Sodium Na	0.46	0.23	1.0	0.31	1.66	0.44
Total	4.46	0.23	10.76	0.31	13.45	0.44
Anions						
Carbonate CO_3	4.2	0.24	12.4	0.29	10.5	0.57
Chloride Cl	1.8	0.06	2.5	0.12	2.84	0.30
Sulphate SO_4	1.35	–	3.48	0.03	11.95	–
Nitrate NO_3	–	–	–	–	1.15	–
Total	7.35	0.30	18.38	0.44	26.44	0.87
Total ions in solution	11.81	0.53	29.14	0.75	39.89	1.31
COST per 1000 gallons	5.22d		9.83d		16.5d	

The table shows the composition of some types of water before and after treatment by Permutit's "Deminrolit" Process. Water similar to a distillate is produced by this process at a fraction of the cost. Where distilled water was too expensive you can afford "Deminrolit" water. The process has been in practical use in Great Britain for over 7 years. Write for technical publication "Distilled Water without Distillation" to

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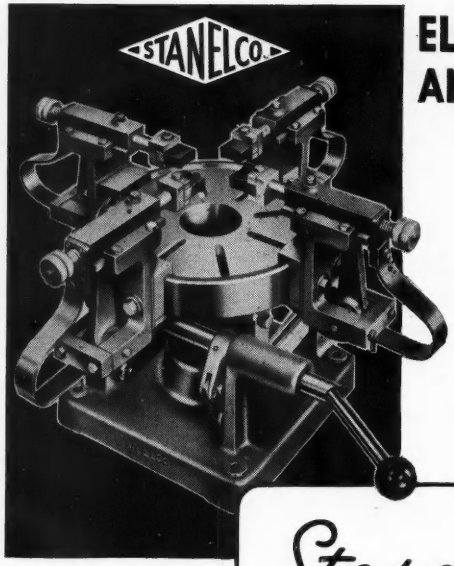

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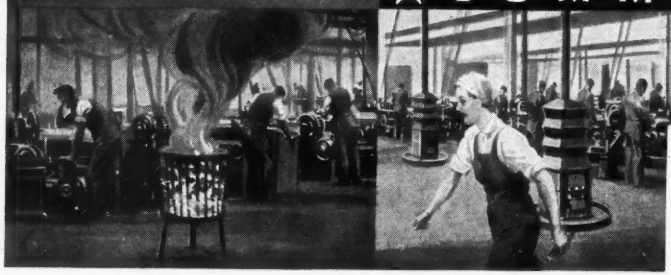

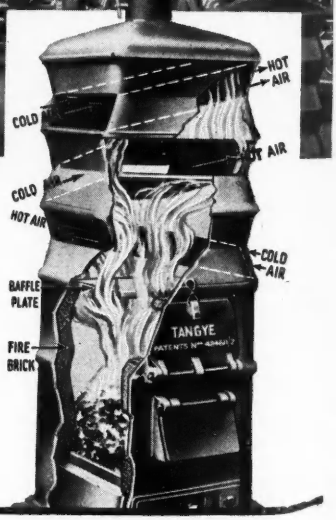

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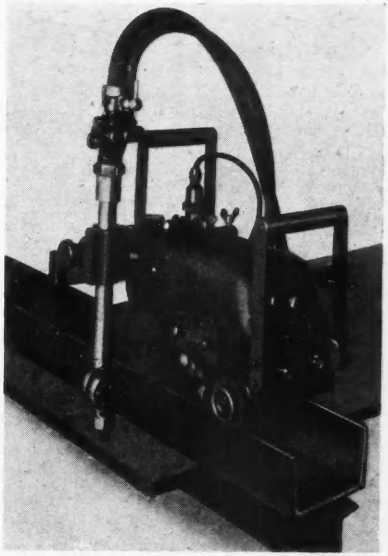

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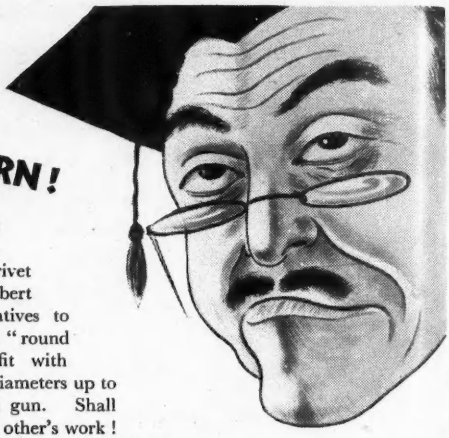

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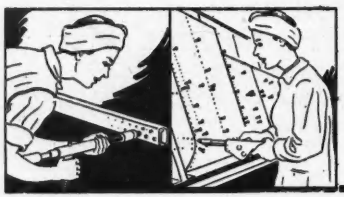

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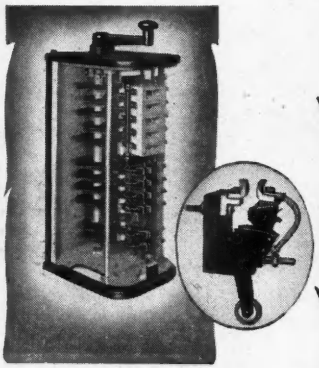

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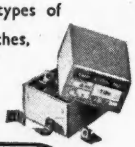

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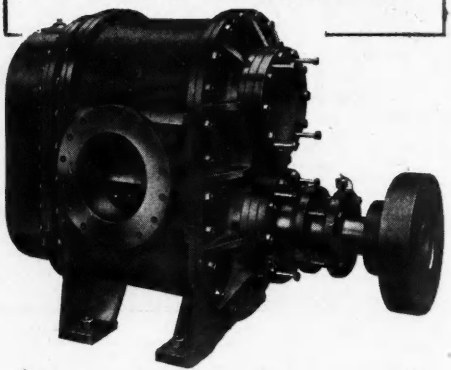

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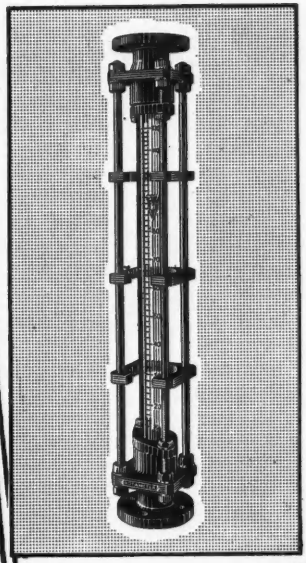

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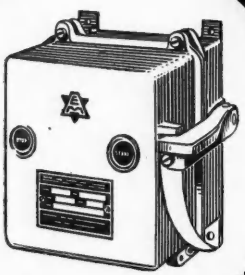

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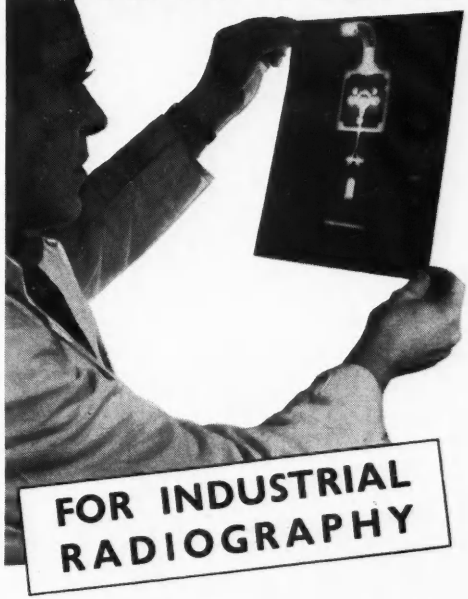

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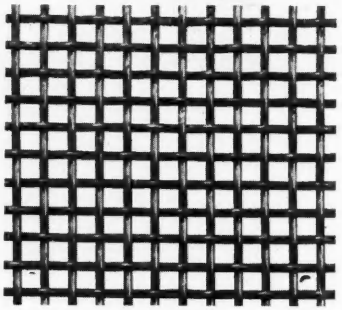

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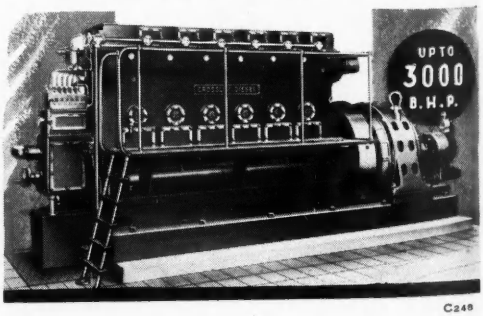

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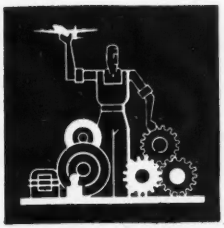

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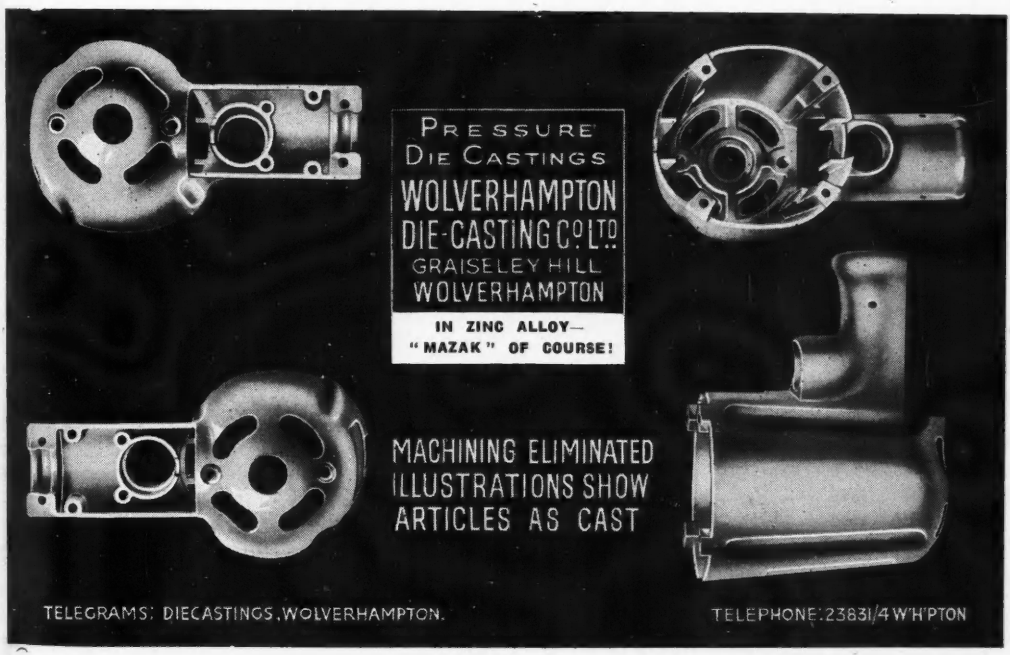

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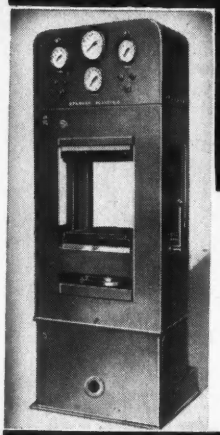

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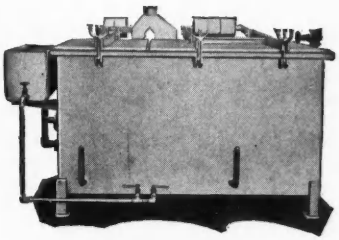

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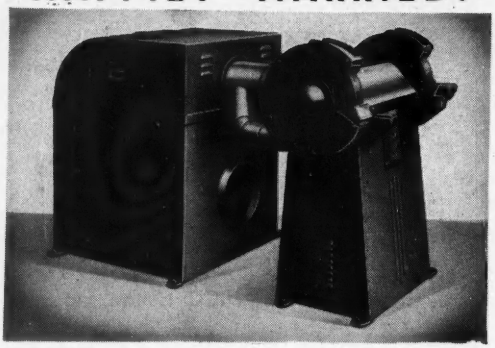

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